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August,1979

Advanced Energy Systems Division

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COMPACT CLOSED CYCLE BRAYTON SYSTEM FEASIBILITY STUDY

FINAL REPORT

Volume I

Research Sponsored by Office of Naval Research
Arlington, Virginia 22217

Under Contract N00014-76-C-0706

Westinghouse Electric Corporation
Advanced Energy Systems Division
P.O. Box 10864
Pittsburgh,Pennsylvania 15236

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20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

This report presents the final results from a three year study which evaluated the feasibility of a closed Brayton cycle power conversion system for compact light weight naval propulsion plants. The results from the first year of this Compact Closed Cycle Brayton System (CCCBS) study were documented in detail in Report Number WAES-TNR-233 of July 1977. next page

This final report includes the detailed results from Years 2 and 3 of this study. It also includes sufficient Year 1 results to provide an understanding

of the total results, but does not include restatements of all of the detailed results derived during Year 1. This approach has been followed in order to fully document the study results while maintaining the report at a size reasonable for the users.

The overall objective of the program was to conduct the analytical study and experimental research required to evaluate and to demonstrate feasibility of a closed Brayton cycle power conversion system for a low volume, light weight marine propulsion plant. Another objective was to insure relevance of power conversion system study results to all candidate applications, including recognition of the various energy sources which the Navy could desire to use in the future.

The scope of the power conversion system for this study excluded specific energy sources, specific loads, and specific heat rejection subsystems. However, consideration was given in the study to the reasonable ranges of conditions for each of these interfaces to insure that the study results meet or exceed the conditions which may be required.

The contracted tasks have been completed as planned and the objective of the study has been fulfilled. As a result of the analytical study and experimental research of this program, the feasibility of a closed Brayton cycle power conversion system for a low volume, light weight naval propulsion powerplant has been shown.

The Compact Closed Cycle Brayton System (CCCBS) program has included derivation of the most stringent representative requirements for the CCCBS power conversion system, consideration of the interfaces with the ship and with other powerplant components which were outside the scope of the system under study, investigation and evaluation of the components which are most critical to feasibility, iterative definition of a reference 52.2 M Pa (17,000 HP) CCCBS design concept; extensive creep-rupture tests of candidate turbine materials in helium and in air at 927°C (1700°F), and overall evaluations and assessments. The results have shown the feasibility and attractive characteristics of a CCCBS and have indicated that no high risk developments or technology breakthroughs are needed for the CCCBS power conversion system.

The results of this program provide a valuable baseline of data for use by the Navy in defining the advanced powerplants which will enhance the capabilities of many types of naval vessels. However, this CCCBS research program has been only one step in the process of actually providing the improved powerplant capabilities that will be needed. This program should be supplemented by vehicle installation and application design work, energy source evaluations and characterizations, additional materials tests, and overall powerplant evaluations and characterizations as next steps towards acquisition of the achievable powerplant gains.

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APPENDICES

1.0 INTRODUCTION

This report presents the final results from a three year study which evaluated the feasibility of a closed Brayton cycle power conversion system for compact light weight naval propulsion plants. The results from the first year of this Compact Closed Cycle Brayton System (CCCBS) study were documented in detail in Reference 1.

This final report includes the detailed results from Years 2 and 3 of this study. It also includes sufficient of the Year 1 results to provide an understanding of the total results, but does not include restatements of all of the detailed results derived during Year 1. This approach has been followed in order to fully document the study results while maintaining the report at a size reasonable for the users.

The overall objective of the program was to conduct the analytical study and experimental research required to evaluate and to demonstrate feasibility of a closed Brayton cycle power conversion system for a low volume, light weight naval propulsion plant. Another objective was to insure relevance of power conversion system study results to all candidate applications, including recognition of the various energy sources which the Navy could desire to use in the future.

In order to accomplish the overall objectives while minimizing program cost and time, the program was carefully planned and conducted to provide the results that were necessary. The overall technical approach of the program was designed to provide the baseline, the continually integrated studies and the integrated evaluations that were necessary to fulfill the program objectives. The functions which comprised the program are shown in Figure 1-1.

The logic applied throughout the program recognized that a basic condition for determination of feasibility is that the evaluations must be conducted in the context of one set of representative requirements which meet or exceed those

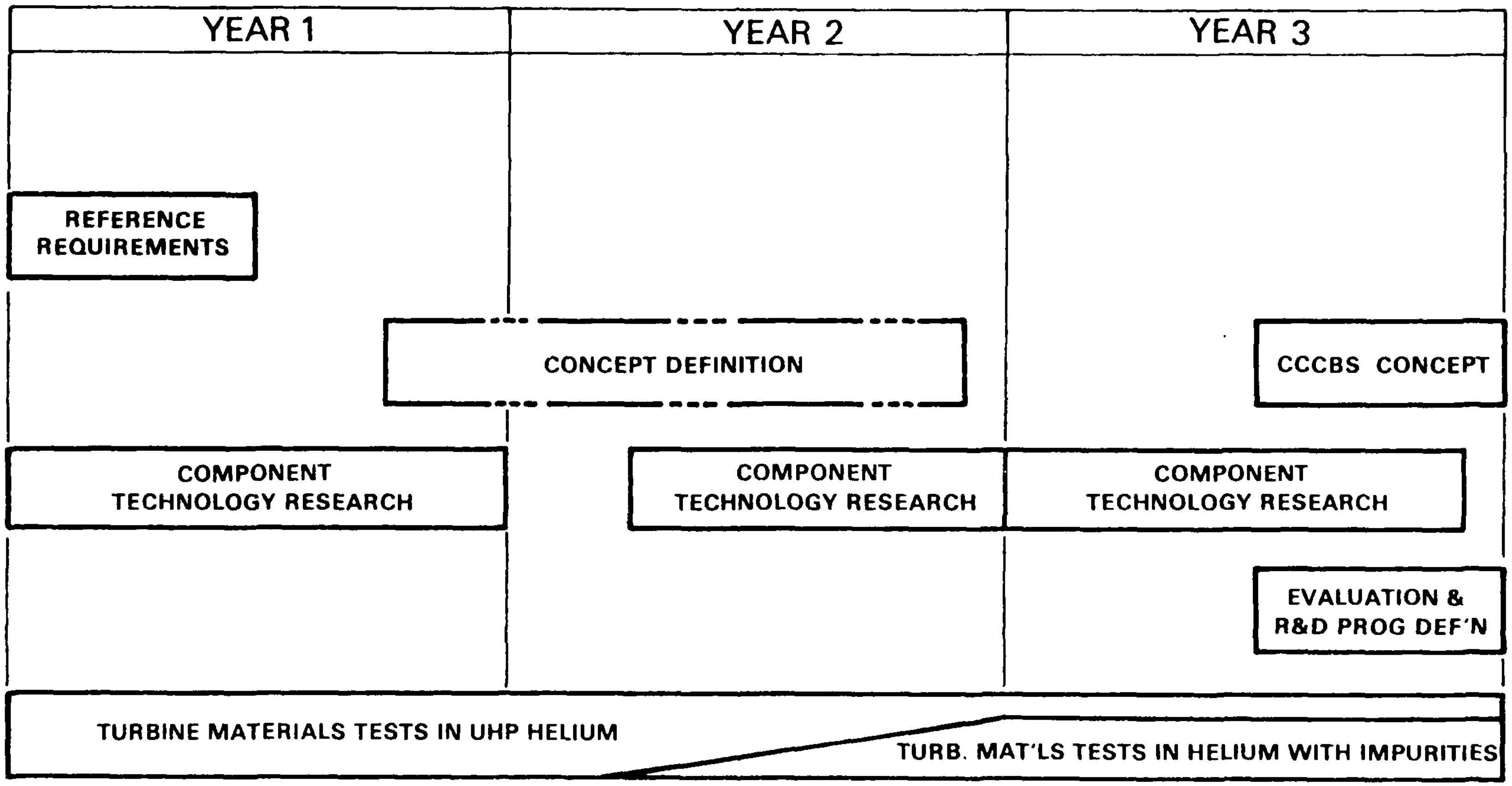

Figure 1-1. CCCBS Research Program

which would be imposed in actual use. One combined set of the most stringent representative top level requirements was therefore determined and used for all program efforts. Because the most stringent requirements were used throughout, the technology evaluations were sufficient for the most demanding case and more than sufficient for others. In this manner, the detailed evaluations of one configuration provided results which meet or exceed the necessary conditions for feasibility assessments of a wide range of possible closed Brayton cycle power conversion system applications.

The scope of the power conversion system for this study, illustrated in Figure 1-2, excluded specific energy sources, specific loads, and specific heat rejection subsystems. However, consideration was given in this study to the reasonable ranges of conditions for each of these interfaces to insure that the study results meet or exceed the conditions which may be required.

The basic reference conceptual design resulting from the study is shown in Figure 1-3. The program objectives required that consideration be given throughout the study to what is required to achieve a compact, light weight powerplant, which in turn requires a very compact power conversion system. However, a compact light weight powerplant requires consideration of the total powerplant as a whole rather than as an assembly of components. Thus, although portions of the total closed Brayton powerplant system were outside the scope defined for detail study, total powerplant considerations were included as necessary to properly define and evaluate the compact closed cycle Brayton power conversion system of this study.

The CCCBS program results have clearly shown the feasibility and practicality of a CCCBS based upon already existing technology. The results also indicate that powerplants based upon a CCCBS will have very attractive characteristics for many naval propulsion applications. It therefore appears to be highly desirable that the next steps be taken so that CCCBS powerplants will be available to enhance the characteristics of US Navy vessels.

This study has been accomplished under the guidance of Mr. J.A. Satkowski, Director of Power Programs in the Material Sciences Division of the Office of

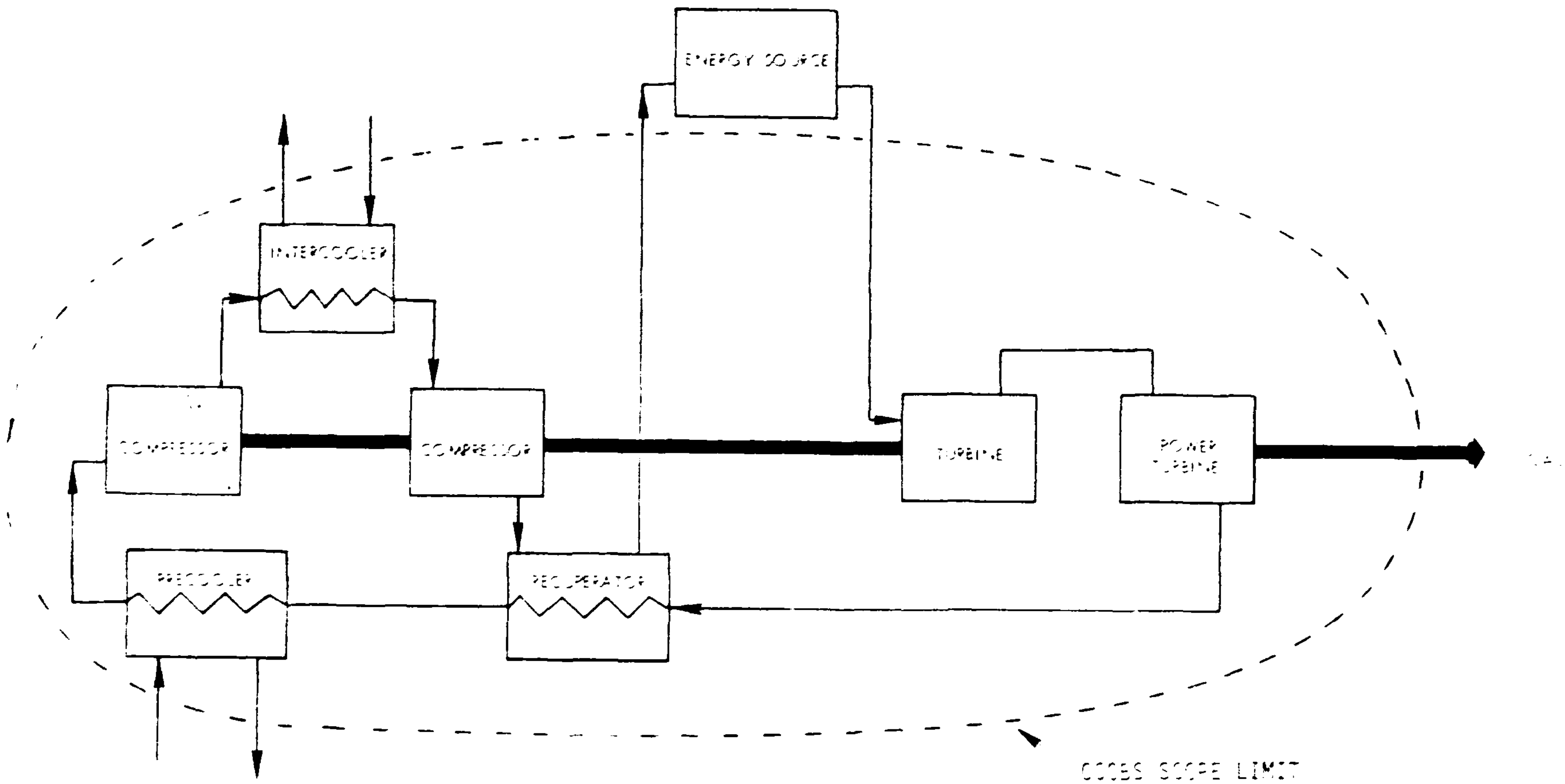

Figure 1-2. System Scope CCGES Study

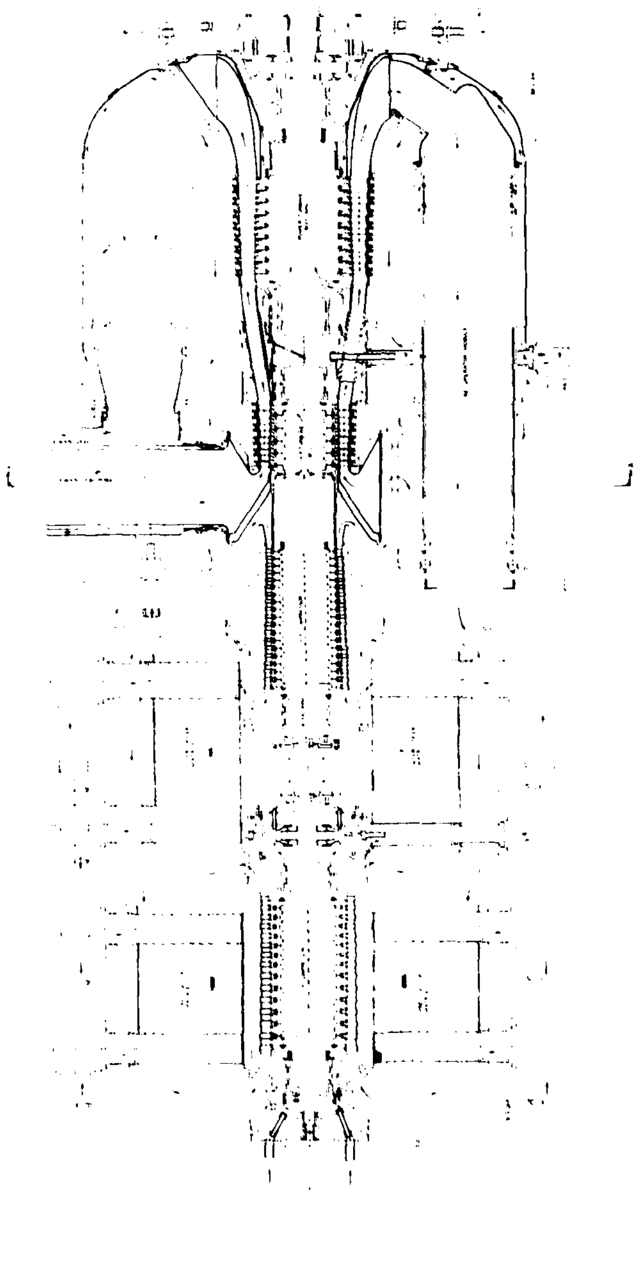

Figure 1-3. Reference CCCBS Design Concept

Naval Research, and LCDR W.R. Seng and Mr. M.K. Ellingsworth who served as the ONR Scientific Officers for this study. The direction and technical input of Mr. Satkowski, LCDR Seng and Mr. Ellingsworth have contributed greatly to the results that have been achieved.

The study was directed by the Westinghouse Advanced Energy Systems Division (AESD) with Mr. R.E. Thompson as Project Manager. The study was accomplished by a team of Westinghouse Advanced Energy Systems Division and the Garrett Corporation's AiResearch Manufacturing Company of Arizona, where Mr. A. Pietsch and Mr. B.B. Heath directed and contributed to the AiResearch efforts. AiResearch provided valuable contributions to the program in various technical areas with particular emphasis upon compact heat exchangers, gas bearings, and rotor dynamics. The program also benefited from technical contributions from other Westinghouse Divisions, particularly from the Combustion Turbine Systems Division and from the Research and Development Center.

1.1 REFERENCES

1. "Compact Closed Cycle Brayton System Feasibility Study, Final Report - Year 1"; Westinghouse Advanced Energy Systems Division, WAES-TNR-233; July 1977.

2.0 SUMMARY AND CONCLUSIONS

2.1 SUMMARY

The contracted tasks have been completed as planned and the objective of the study has been fulfilled. As a result of the analytical study and experimental research of this program, the feasibility of a closed Brayton cycle power conversion system for a low volume, light weight naval propulsion powerplant has been shown.

The Compact Closed Cycle Brayton System (CCCBS) program has included derivation of the most stringent representative requirements for the CCCBS power conversion system, consideration of the interfaces with the ship and with other powerplant components which were outside the scope of the system under study, investigation and evaluation of the components which are most critical to feasibility, iterative definition of a reference 70,000 HP CCCBS design concept, extensive creep-rupture tests of candidate turbine materials above the expected turbine inlet material temperature, and overall evaluations and assessments. The results have shown the feasibility and attractive characteristics of a CCCBS and have indicated that no high risk developments or technology breakthroughs are needed for the CCCBS power conversion system.

The results of this program provide a valuable baseline of data for use by the Navy in defining the advanced powerplants which will enhance the capabilities of many types of naval vessels. However, this CCCBS research program has been only one step in the process of actually providing the improved powerplant capabilities that will be needed. This program should be supplemented by vehicle installation and application design work, energy source evaluations and characterizations, additional materials tests, and overall powerplant evaluations and characterizations as next steps towards acquisition of the achievable powerplant gains.

2.2 CONCLUSIONS

2.2.1 GENERAL CONCLUSIONS

a. Feasibility and practicality of a Compact Closed Cycle Brayton System have been demonstrated through the detailed concept design and evaluation results of these program efforts and have been supported by the favorable results of the creep-rupture materials tests.

b. These evaluations have been accomplished in the context of representative most stringent powerplant requirements. Therefore, the conclusion of CCCBS feasibility is applicable to a wide range of naval vessel applications and energy sources.

c. No high risk development or major technology development have been identified to be needed. Instead, the development program required is that of reduction to practice.

d. In addition to compactness and light weight, the CCCBS has been determined to have other desirable characteristics of importance in naval propulsion applications. These include, among others:

- High efficiency (low SFC) at rated conditions with the capability of even higher efficiency through increased recuperation if determined to be needed.
- Much less reduction in efficiency (increase in SFC) at partial power than is characteristic of open cycle gas turbine systems.
- Inherent quietness
- Flexibility in powerplant location in the vessel
- Potential for long lifetime
- Compatibility with a variety of energy sources

e. A design concept was derived for a compact, integrated assembly, 52.2 MW (70,000 horsepower) power conversion system (PCS) which fulfills the needs of a low volume, light weight propulsion powerplant. Evaluations have shown the design concept to be capable of fulfilling all expected Navy design requirements for such a power conversion system.

f. Evaluations of the most critical components and analyses of system transient conditions have supported the validity of the design concept and the overall conclusion of feasibility.

g. The reference conceptual design 52.2 MW (70,000 HP) CCCBS power conversion assembly has an outside diameter of 2.33 m (92 in.) and a length of 5.49 m (216 in.). The weight of the unit is estimated to be 42,519 Kg (93,755 lb) which is a specific weight of 0.81 Kg/Kw (1.34 lb/HP).

h. The reference design concept has been evaluated to identify and quantify the effects of changes in the reference requirements.

i. Creep-rupture testing of candidate first-stage turbine materials was conducted at 1700°F. Testing of five materials was conducted in three different environments; air, ultra-high purity helium, and helium containing controlled amounts of impurities simulating a CCCBS working fluid. The results in the helium environments were within the expected statistical data band when compared to the results obtained in air. The materials tests are therefore considered to be an important indication that suitable materials can be selected for the CCCBS working fluid environment.

j. The development program necessary for a CCCBS has been assessed. A reasonably paced program has been estimated to require 7-1/4 years to achieve a MIL Specification qualified system.

k. The results of this CCCBS research program should be supplemented by vehicle installation and application design work, energy source evaluations and characterizations, additional materials tests, and overall powerplant evaluations and characterizations.

2.2.2 SYSTEM CONCLUSIONS

a. Representative most stringent requirements were defined and used as a basis for evaluations of feasibility. Among the reference requirements were a unit output of 52.2 MW (70,000 HP), turbine inlet gas temperature of 927°C (1700°F), and use of helium as the working fluid.

b. A recuperated and intercooled cycle was shown to be most attractive for anticipated Navy requirements, and also provided the most severe case for evaluations of feasibility.

c. For compactness and light weight of the CCCBS, helium has been determined to be a desirable working fluid.

d. A design concept has been defined which provides the necessary compactness. The base case design was derived to interface with the superconducting generator of an electric power transmission system. A different power turbine configuration was defined to provide 3600 rpm shaft power output to a conventional reduction gear assembly.

e. An integrated assembly was determined to be necessary for the compactness requirements imposed by vehicles upon the powerplant. However, the evaluations of this study have not only shown such an assembly to be practical, but have also shown that the integrated assembly has many other advantages including:

- Reduced susceptibility to leakage
- Inherent shock resistance
- Reduced effects of thermal expansion and contraction during startup and shutdown, thereby enhancing lifetime and reliability
- Enhanced reliability (elimination of interconnecting piping and individual component casings reduces failure possibilities)
- Mounting is simplified and effects on the powerplant of flexibility of the ship are reduced
- Small quantity of working fluid required allows practical inventory control and provides a rapid transient response capability
- Inherent quietness

f. The reference conceptual design has been derived such as to be capable of being highly reliable and includes provisions for maintenance consistent with expected shipboard maintenance facilities and personnel capabilities. Very little maintenance is expected to be required because of the design to minimize the effects of individual component failures and because of the benign working fluid environment

g. Very long lifetime of the power conversion system is expected to be readily attainable, possibly approaching the lifetime of the ship. Some of the factors enhancing power conversion system lifetime are:

- Inert working fluid
- No products of combustion in the working fluid
- No salt ingestion into the working fluid
- Relative uniformity of temperatures over the operating range

h. Evaluations have shown that the integrated assembly reference design is capable of fulfilling the Navy requirements for shock resistance.

i. No major difficulties are foreseen in the derivation of Military Specifications for the CCCBS. The CCCBS power conversion system as a whole and its components have been derived such as to be responsive to the intent of the most nearly appropriate current Military Specifications.

j. CCCBS dimensions and weights have been estimated for powerplant and power transmission system applications, as shown in Table 2-1. Compactness and light weight are indicated by these characteristics.

TABLE 2-1

CCCBS POWER CONVERSION SYSTEM SIZE AND WEIGHT
52.2 MW (70,000 HP)

	9000 rpm Power Turbine	3600 rpm Power Turbine
Outside Diameter	2.33 m (92 in.)	2.33 m (92 in.)
Overall Length	5.49 m (216 in.)	6.50 m (256 in.)
Weight*		
- Casing for Two Unit Operation	42,519 Kg (93,755 lb)	50,233 Kg (110,763 lb)
- Casing for One Unit Operation	34,060 Kg (75,103 lb)	40,886 Kg (90,153 lb)
Specific Weight*		
- Casing for Two Unit Operation	0.81 Kg/Kw (1.34 lb/hp)	0.96 Kg/Kw (1.58 lb/hp)
- Casing for One Unit Operation	0.65 Kg/Kw (1.07 lb/hp)	0.78 Kg/Kw (1.29 lb/hp)

*For two parallel CCCBS operation from a single energy source, the CCCBS outer casing must be sized to withstand full system pressure while for single unit operation the casing need only be sized to withstand the lower local pressure.

k. The effects of changes in design turbine inlet temperature, design pressure level, and design power output have been assessed. It has been determined that the feasibility and performance of the CCCBS is not seriously impacted by changes in these design parameters. Quantitative estimates of the effects of these parameters have been defined.

l. Analyses of normal and malfunction transients have shown the CCCBS to be very controllable, to provide rapid transient response, and to be capable of withstanding the most severe malfunction transients that have been identified.

2.2.3 TURBOMACHINERY CONCLUSIONS

a. Feasibility and practicality of turbomachinery with a design turbine inlet temperature of 927°C (1700°F) in helium with uncooled turbine blades and simply cooled turbine discs has been shown.

b. Critical turbomachinery components (e.g., first turbine stage, last power turbine stage, first compressor stage, and bearings) were defined and evaluated in detail. These evaluations have quantified turbomachinery charcteristics and have provided input to evaluations of feasibility of the turbomachinery.

c. Compressor efficiency of 85% and turbine efficiency of 90% at rated conditions has been evaluated as being realistically achievable.

d. Both gas bearings and oil lubricated tilting pad bearings have been evaluated and determined to be capable of fulfilling the bearing requirements. Because of their numerous advantages to the overall system, gas bearings were selected for the reference concept design.

e. Gas bearings were evaluated in detail and hydrostatic gas bearings were determined to be capable of fulfilling all requirements while providing very desirable characteristics. Appropriate hydrostatic gas bearing designs were derived for each journal and thrust bearing.

f. Capability of the turbomachinery for qualification in accordance with the requirements of MIL-S-901C (Navy) was analytically evaluated and determined to be satisfactory.

g. A reference power turbine design concept was defined at 9000 rpm for drive of a superconducting generator. Various alternates for provision of 3600 rpm shaft power and for positive shaft sealing were also evaluated and an alternate 3600 rpm shaft power output turbine design concept defined and characterized.

h. Overspeed protection of the power turbine, in the event of complete loss of load, was evaluated and several suitable alternatives defined to provide assured overspeed protection.

k. The effects of changes in design turbine inlet temperature, design pressure level, and design power output have been assessed. It has been determined that the feasibility and performance of the CCCBS is not seriously impacted by changes in these design parameters. Quantitative estimates of the effects of these parameters have been defined.

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h. Overspeed protection of the power turbine, in the event of complete loss of load, was evaluated and several suitable alternatives defined to provide assured overspeed protection.

2.2.4 HEAT EXCHANGER CONCLUSIONS

a. Various types of compact gas-to-liquid (precooler and intercooler) and gas-to-gas (recuperator) heat exchangers were evaluated in the context of the CCCBS requirements. For each heat exchanger, at least one suitable alternative was determined to exist, thus providing enhanced assurance of feasibility.

b. In general, the compact heat exchangers necessary were determined to be within current aircraft-type heat exchanger state-of-the-art and to require no technology breakthroughs. The needs for compactness and reliability do require that the heat exchangers be well engineered and manufactured with appropriate product assurance throughout.

c. Helical tube and fin precoolers and intercoolers were defined in detail for the CCCBS reference design concept. These heat exchangers were determined to be fully suitable for the CCCBS.

d. Small tube recuperator modules were defined in detail for the CCCBS reference design concept. These heat exchangers were determined to be fully suitable for the CCCBS.

2.2.5 MATERIALS TESTING CONCLUSIONS

a. Creep-rupture materials tests at 927°C (1700°F) in ultra-high-purity (UHP) helium, in helium containing controlled levels of impurities simulating the expected CCCBS working fluid environment, and in air have provided important data showing that suitable materials can be selected for a CCCBS.

b. Five candidate materials were tested (Alloy 713LC, IN 100, MAR-M509, MA 754, and the refractory TZM). Total testing included 76,583 hours in UHP helium, 17,256 hours in helium containing controlled levels of impurities, and 40,035 hours in air.

c. No deleterious effects of the UHP helium or of the helium with impurities environments on the creep-rupture behavior of the materials tested have been observed.

d. Although the materials tests to date provide sufficient data to support the feasibility of a CCCBS, additional materials tests are required as part of the next steps in providing a CCCBS. Additional creep-rupture testing over a range of temperatures and of other materials should be accomplished. The effects, if any, of the working fluid environment on fracture mechanic properties and on low and high cycle fatigue properties should be determined by tests.

e. Some alloy optimization research should be directed specifically toward inert gas environment applications so as to provide enhanced material capabilities by taking advantage of the benign working fluid environment.

3.0 EVALUATION OF FEASIBILITY

The overall objective of this study is:

> "Conduct the analytical study and experimental research required to evaluate and to demonstrate feasibility of a closed cycle Brayton power conversion system for a low volume, light weight marine propulsion plant."

This objective has guided the study from inception through completion and has determined the logic for evaluation of feasibility and has guided the individual tasks during their accomplishment.

The scope of the power conversion system for this study, illustrated in Figure 3-1, was established to include the components which are necessary to convert the input energy to output power. Interfaces with other systems which will vary from application to application were to be considered, but were not specifically included as part of the power conversion system. Thus, specific energy sources, specific loads and their transmission systems, and specific heat rejection systems were not included but the interfaces with those powerplant components, and the ranges of interface conditions, were considered to insure that the study results are valid.

Feasibility of a Compact Closed Cycle Brayton System (CCCBS) has been demonstrated by the results of the design, analytical and materials testing activities that were accomplished under the contract. This section presents the results of the evaluations of feasibility and the overall characteristics of the CCCBS. Other sections of the report document the detailed technical investigation results that provided inputs to the feasibility evaluations.

3.1 LOGIC FOR EVALUATION

There is an inherent difficulty in any evaluation of feasibility and that difficulty results from the fact that evaluation and conclusions regarding feasibility necessarily have to be accomplished in the context of known or assumed requirements. It is all too easy for evaluations to be accomplished within

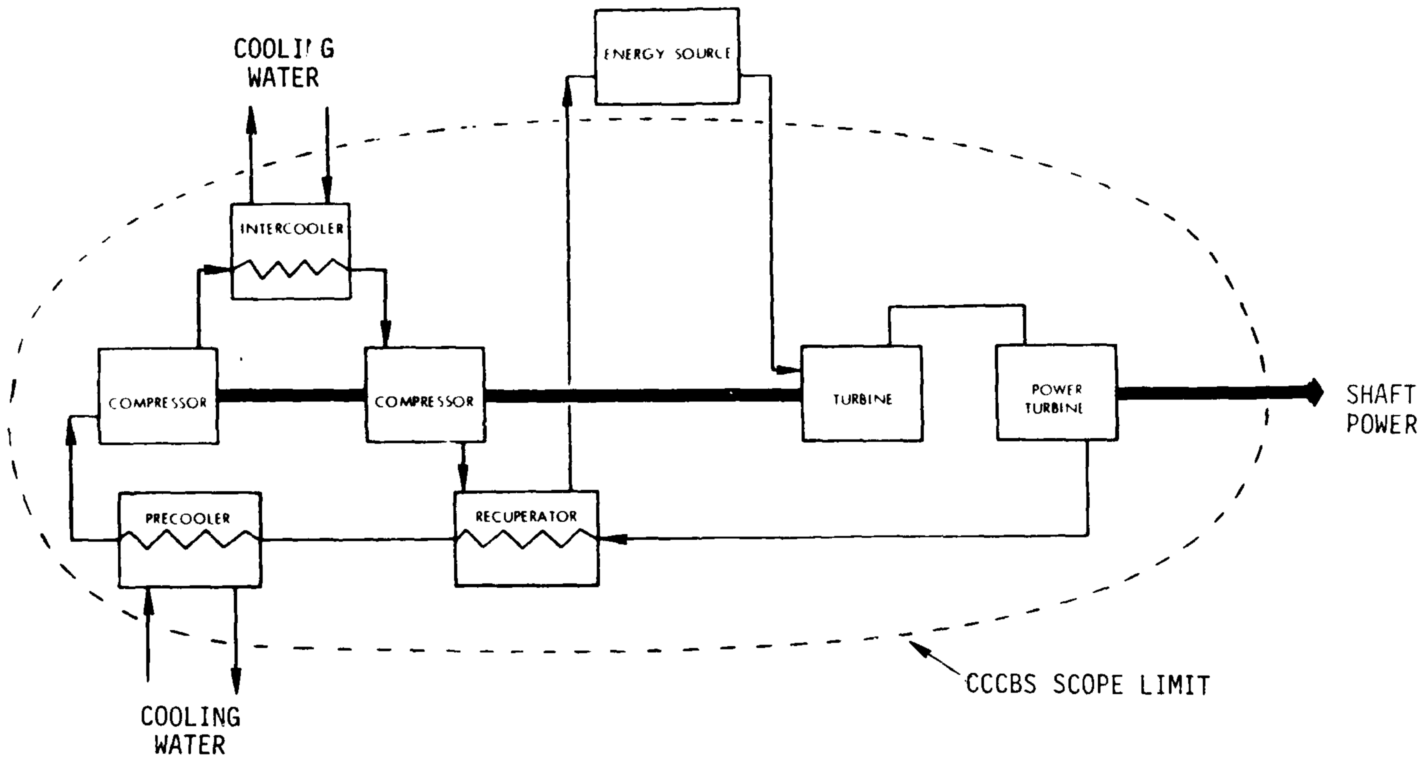


Figure 3-1. System Scope of CCCBS Study

the context of such specialized requirements in a manner such that the results are useful only for the specific case examined and have little general usefulness. Since the purpose of this study is to provide results which would be of maximum general usefulness to the Navy and to other agencies, very careful attention was given to establishment of the bases upon which feasibility would be evaluated and to the methodology of the study.

In order to make the results generally applicable, two different approaches could be followed. One approach is to examine each of the possible applications and to evaluate a power conversion system (PCS) for each application. This approach is very costly in terms of both money and time. Another approach is to examine each of the possible alternate applications and determine the most stringent requirements which can realistically be expected to be imposed on a power conversion system. Since the objective of the study is to evaluate feasibility, the latter approach was followed so as to show feasibility for a wide range of applications at reasonable cost. For instance, there are a number of different energy sources which may be in use or desired for use by the Navy in the 1990s, each of which can place different requirements and/or limitations on the power conversion system. Early in this study, the various energy sources were considered and the resulting most stringent requirements and limitations were used in the study as the energy source requirements on the power conversion system.

The generalized logic of the three year program is shown in Figure 3-2. Creep-rupture testing of candidate materials for first stage turbine blades was conducted in parallel with these efforts throughout the program. The logic recognizes that it is basic to determination of feasibility that the evaluations be conducted in the context of top level requirements which are at least as severe as those which would be placed upon the power conversion system in actual use. All efforts were accomplished in the context of these same top level requirements. The overall technical approach was designed to provide the requirements baseline, continually integrated design studies and critical component technology research, and integrated evaluations that were necessary to fulfill the program objective.

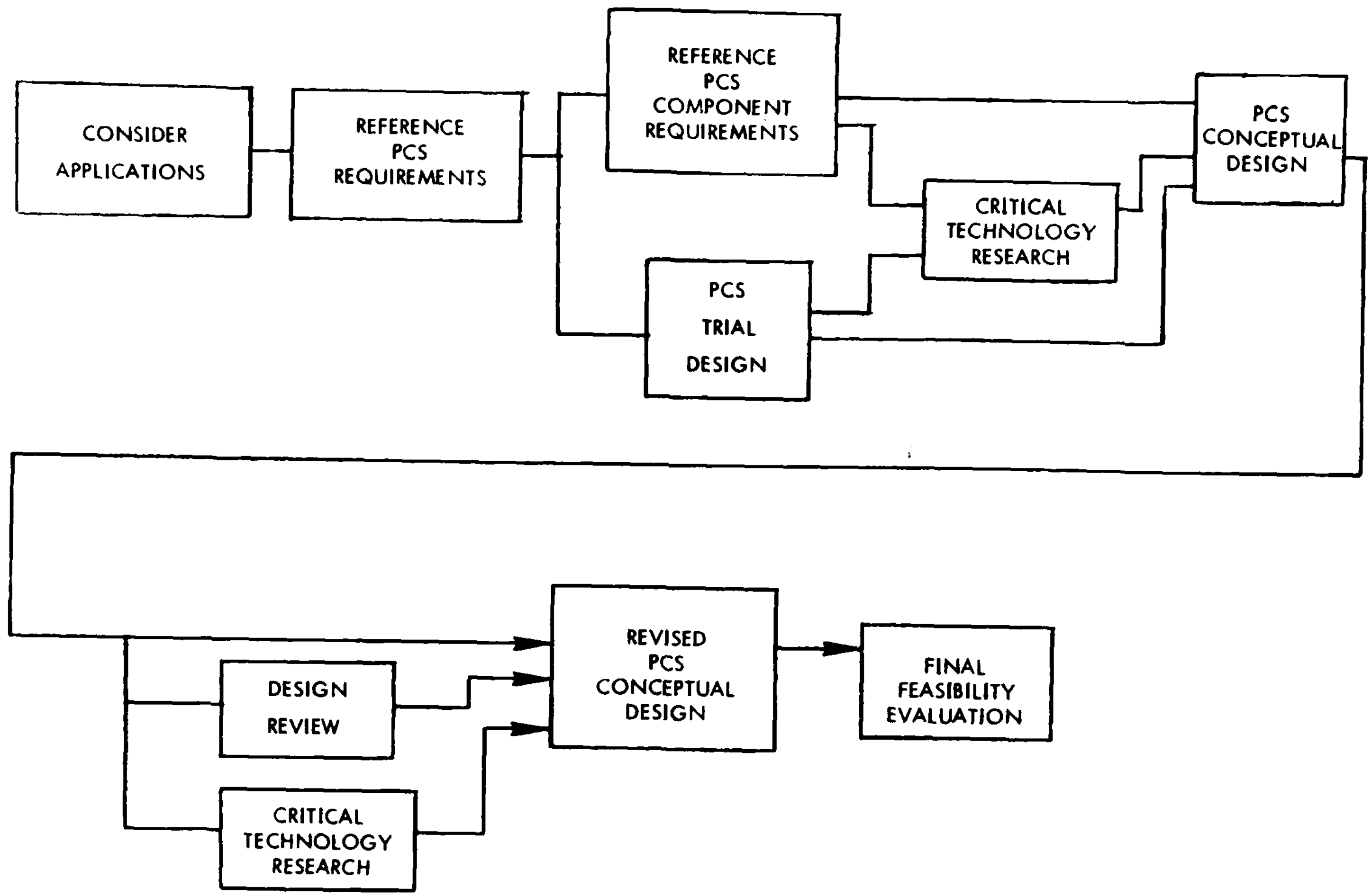


Figure 3-2. Generalized Logic of Program

The final evaluations of feasibility have essentially assumed that no more than a low risk in development is acceptable. This assumption has been made because of the judgment that feasibility of a new concept or, as in this case, of a new use of an old concept must be conservatively assessed if the conclusions are to be accepted and used.

3.2 REQUIREMENTS

3.2.1 TOP LEVEL REQUIREMENTS

Reference Application - Surface Effect Ship

The needs of various types of naval vessels were considered. Because the study is directed toward compact, light weight powerplants, emphasis was placed upon the advanced vessel types for which powerplant compactness and/or light weight are important parameters. Surface Effect Ships (SES), hydrofoil and advanced displacement ships are the major types of vessels given emphasis.

The limitations placed upon the powerplant by the vessel are most demanding for the SES application. This particular application tends to be very demanding with regard to space, weight, accessibility and load variability. It also requires consideration of power conversion system matching with electric or mechanical power transmission systems. Indications are that powerplants which fulfill the requirements of SES applications would also meet or exceed the requirements of other applications. Therefore, the SES was selected as the reference application with the recognition that the CCCBS design studies must not include features which would exclude other applications.

Unit Power Conversion Assembly Output - 52.2 MW (70,000 SHP)

Studies have indicated that unit power for the various applications can be expected to range from 54.9 MW (20,000 SHP) to 74.6 MW (100,000 SHP) with the more likely unit power required in the range of 44.7 MW (60,000 SHP) to 74.6 MW (100,000 SHP). The needs of the high performance ships appear to favor unit powers toward the high end of the range. To make the results of this study most generally applicable, it was judged that the unit power selected for the study should be appropriate for the high power range, but also should be reasonably scalable to lower powers. It was determined that a unit power level of 52.2 MW (70,000 SHP) would fulfill these criteria and would also provide the benefit to this study of direct infusion of the results or prior Westinghouse funded studies.

In addition to unit power, it was also determined that the CCCBS power conversion assembly must be capable of operation in the configuration of two power

conversion assemblies coupled in parallel to a single energy source. Such a configuration has been found to be needed for some installations such as large SES. Because such an installation imposes more stringent conditions on the CCCBS evaluations, this configuration capability was established as a requirement.

It was also established that, for general applicability, the CCCBS evaluations must include provisions to accept both fixed and variable speed loads.

Specific Weight of Power Conversion Systme - 1.2 Kg/Kw (2 lb/SHP)

The most restrictive weight limitations are placed by the SES application. On the order of 9.1 Kg/Kw (15 lb/SHP) has been determined to be a reasonable value for the total weight of powerplant and fuel. Of this amount, on the order of 1.2 Kg/Kw (2 lb/SHP) is a reasonable, but stringent, upper limit for the CCCBS power conversion system. However, it must be recognized that the usefulness of the CCCBS is enhanced if the specific weight is lower than this value and the study should emphasize attainment of lower weight.

Lifetime - 10,000 Equivalent Full Power Hours (40,000 Operating Hours)

The most stringent lifetime requirement was determined through consideration of the need for the CCCBS to be compatible with any of the various energy sources that the Navy may desire to use. Application with a nuclear energy source would place the most stringent lifetime requirement upon the CCCBS power conversion assembly because of the long reactor lifetimes that are possible and the desire for the power conversion assembly to have a comparable lifetime.

Studies have shown that reactor lifetimes of 10,000 Equivalent Full Power Hours (EFPH) can be planned for naval applications. The duty cycle is, of course, a function of the ship and its normal mission. Since some applications can have a normal duty cycle approximately 25 percent, the CCCBS evaluations must also consider that the total operating time to achieve 10,000 EFPH may be as high as 40,000 hours.

Shock - MIL-S-901C (Navy)

The system shall be suitable for qualification in accordance with the requirements of MIL-S-901C (Navy).

Heat Rejection - To Seawater at 29°C (85°F)

All candidate applications can utilize seawater for heat rejection, with the exception of Air Cushion Vehicles (ACV). However, the ACV is considered to be a special case with the least likelihood of actual implementation with a CCCBS. Therefore, heat rejection shall be considered to be to seawater. For these feasibility evaluations, a conservatively high seawater temperature of 29°C (85°F) shall be used as the most stringent requirement.

3.2.2 SECOND LEVEL REQUIREMENTS

To amplify the top level requirements discussed above, a set of second level requirements was established at the component level. Some of these requirements are reviewd below with a brief description of the trial design given as an aid in specifying the component requirements.

Integrated Assembly

To achieve the compactness needed for low specific weight, an integrated assembly of the turbomachinery and heat exchangers is needed.

The flow path of the working gas in an integrated assembly is illustrated in simplified form in Figure 3-3. High temperature gas from the heat source enters the module through the inner pipe of a concentric duct and enters the high pressure compressors which are of the axial type for high efficiency. After leaving the high pressure turbine, the gas flows to the low pressure turbine (power turbine) where the work necessary to drive the load is extracted. A free power turbine was specified because of the requirement to be able to interface with either fixed or variable speed loads. The gas then enters the low pressure side of the recuperator and is subsequently cooled in the precooler. The gas is then compressed in the low pressure compressor, cooled in the intercooler and further compressed in the high pressure side of the recuperator, where it is heated by the energy extracted from the low pressure side gas. The heated gas then passes into a collector manifold from which it is piped through two check valves in series to the outer annulus of the concentric duct to the heat source.

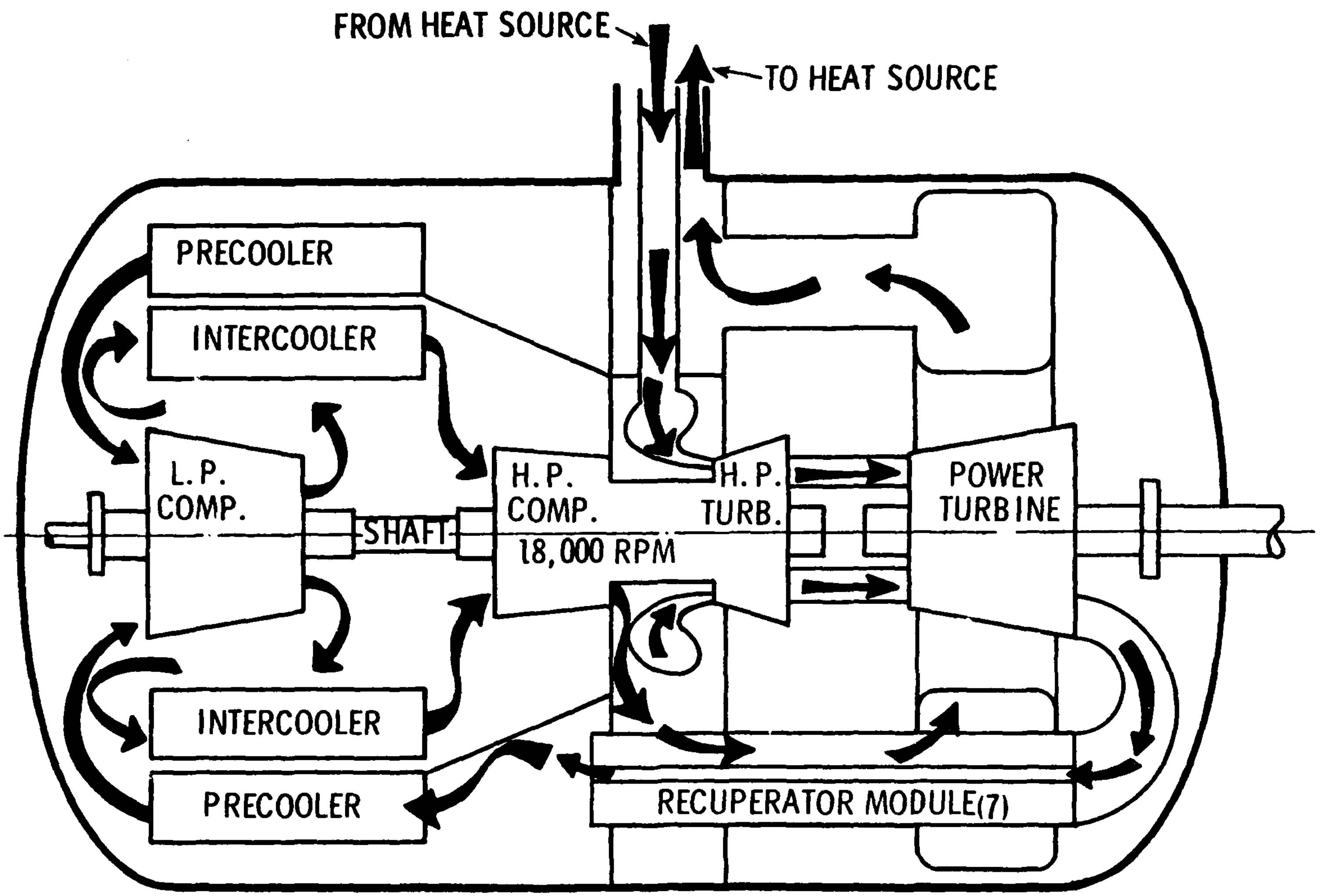


Figure 3-3. Helium Working Gas Flow Path

Working Fluid and Pressure Level - Helium, 10.3M Pa (1500 psia) Turbine Inlet Pressure

A number of investigators have addressed the question whether it is possible, by mixing a light gas, such as helium with a heavy gas, to arrive at a working medium for closed cycle Brayton systems which has the properties which would allow optimizing with respect to cost and layout of the components. The Office of Naval Research has supported heat transfer research reported in References 1 and 2, which indicate that the different heat transfer of these low Prandtl Number gas mixtures had not previously been adequately accounted for. Using the heat transfer correlation recommended in Reference 2, the relative heat exchanger volume ratios for pure gases, air, and binary mixtures were determined and are shown in Figure 3-4 with the same gas or gas mixture on both the tube and shell side (typical recuperator). The use of any heavier molecular weight gas or gas mixture in place of helium in the CCCBS would require increased heat exchanger volume and therefore increased wight. Since the CCCBS requirements include compactness, light weight and compatibility with both fossil fueled and gas cooled nuclear energy sources, helium is the choice for working fluid.

The requirement for compactness dictates the use of as high a system pressure as is practical. Considerations of the compressor, turbine, seals, and of prior studies during the Marine Gas Cooled Reactor (MGCR) closed Brayton cycle turbomachinery program (Reference 3) indicate that 10.3M Pa (1500 psia) is a relatively high, yet achievable, turbine inlet pressure level. Therefore, 10.3M Pa (1500 psia) was established for the studies.

Turbine Inlet Temperature - 927°C (1700°F)

Special attention was given to the selection of the turbine inlet temperature (TIT). Cycle efficiency considerations dictate the use of the highest practical temperature. Based on expected materials properties, it was concluded that a design turbine inlet temperature on the order of 927°C (1700°F) pushes but does not exceed the state-of-the-art for uncooled turbine blades and stators in inert gases. A gas cooled nuclear heat source could supply helium at 927°C (1700°F). Based on the results of the state-of-the-art survey and reasonable design practices using superalloys, fossil energy source heat exchangers can

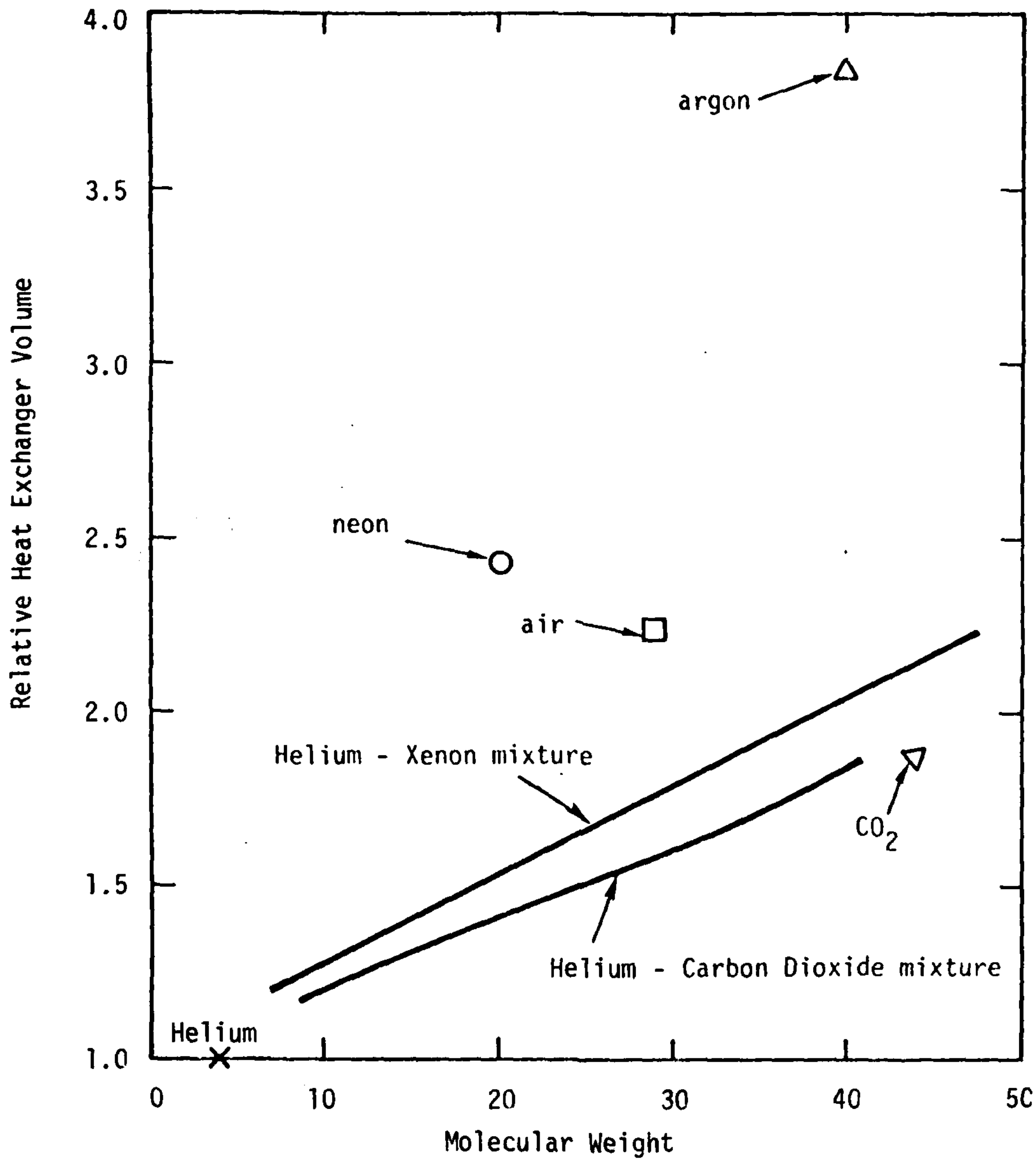


Figure 3-4. *Relative Heat Exchanger Volume for Pure Gases, Air, and Binary Mixtures* (Same gas on both tube and shell side)

be expected to produce working fluid gas temperatures less than 927°C (1700°F). Therefore, a reference design TIT of 927°C (1700°F) was selected as the most stringent requirement. The latitude exists to reduce the turbine inlet temperature for extended off-design point (part-power) operations.

State Points and Components Efficiencies

The temperatures, pressures, helium flow rates, and component efficiencies used for initial component studies were as specified in Table 3-1. These values were derived from work on the trial design and parametric studies and formed a basis for the initial component requirements.

TABLE 3-1

STATE POINTS AND COMPONENT EFFICIENCIES*

	Temperatures - °C (°F) Inlet	/ Outlet	Inlet Pressure - M Pa (psia)	Component Efficiency - %
Energy Source	429 (805)	943 (1730)	10.9 (1580)	-
Turbine	927 (1700)	690 (1275)	10.3 (1500)	90
Power Turbine	690 (1275)	516 (960)	5.8 (840)	90
Recuperator	502 (935)	224 (435)	3.2 (470)	80
	138 (280)	432 (810)	11.0 (1600)	
Precooler	224 (435)	38 (100)	3.2 (460)	98
Compressor	38 (100)	152 (305)	3.1 (450)	85
	38 (100)	138 (280)	6.0 (875)	

*Nominal 61.2 kg/sec 135 (lb/sec) helium flow rate, 150 MW(t) heat source 52.2MW (70,000 HP) Output.

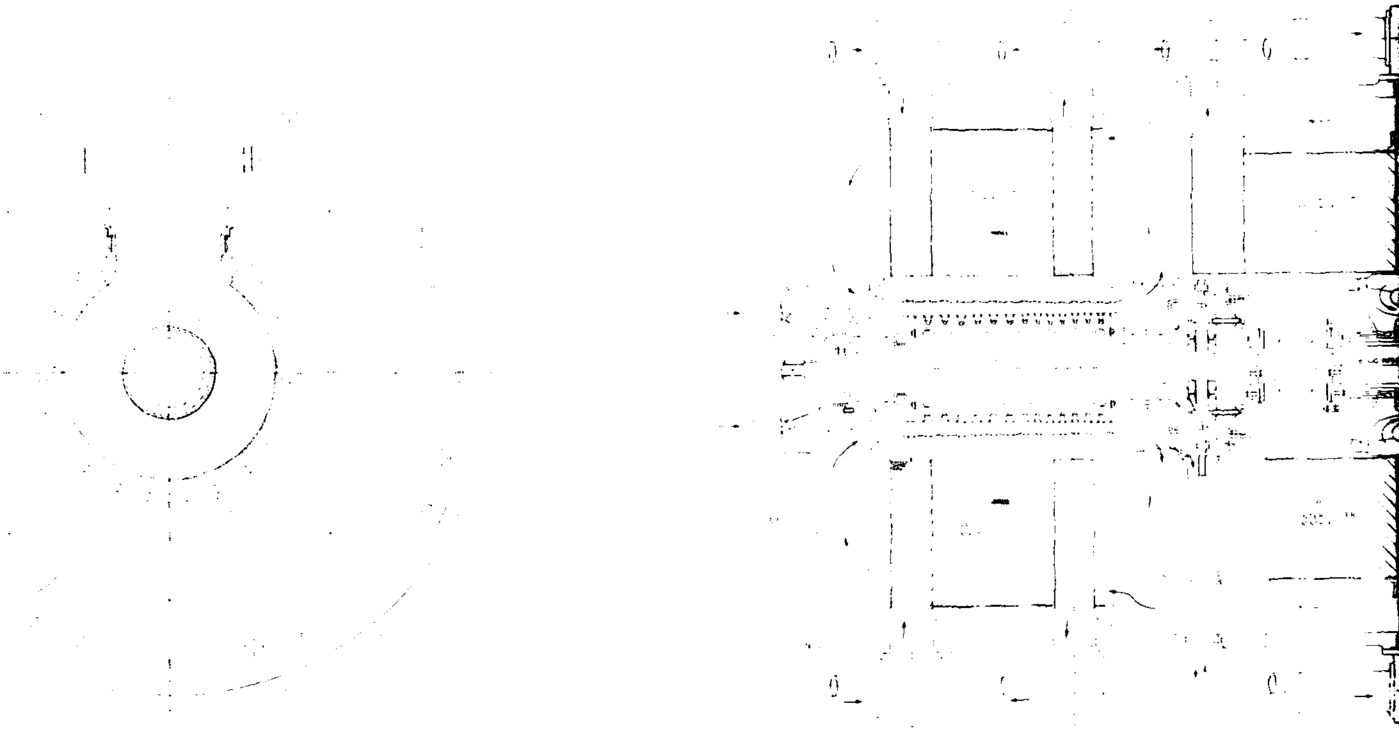
NOTE: These data used only for initial component evaluations.

3.3 DESIGN CONCEPT DESCRIPTION

The power conversion system is designed as an integrated package whose major components can be individually assembled and checked and which can be completely assembled and checked out as an assembly. The design layout is illustrated in Figure 3-5. The components include low and high pressure compressors and high pressure turbine rotating at 18,000 RPM, low pressure free power turbine rotating at 9,000 RPM (at full power), recuperator, precooler and intercooler. (A design variation with a 3600 RPM power turbine was also defined and evaluated during the study). Trade studies between candidate types of heat exchangers and bearings were performed as input to the concept definition. The design concept incorporates the results of those trade studies.

All of the major components of the power conversion assembly are contained within a cylindrical pressure vessel made of low alloy steel, capable of accommodating the maximum system pressure. Although local pressures throughout most of the power conversion assembly are below maximum system pressure under normal operating conditions, operation with one of two parallel units shut down dictates casing design conditions because pressure throughout the inoperative unit tends to approach full system pressure as a result of leakage through the shut-off valves. Consideration has been given to the need for access to the turbomachinery for maintenance through providing for its easy removal and reinsertion into the power conversion assembly.

The pressure vessel is made in three sections; a center frame structure and forward and rear sections. The three sections are joined together by bolted flanges which are sealed by elastomer or metal O-rings, or alternatively, seal welded. The maximum temperature of the pressure vessel is approximately 204°C (400°F). The forward section of the pressure vessel accommodates the intercooler and precooler in an annular arrangement around the turbomachinery. Both units employ a crossflow configuration using finned water cooled tubing arranged in a helical fashion. The tubing is headered in four equally spaced radial pipes at each end of the matrix. A "four-start" arrangement of the tubing provides compatibility with the header pipes. Cooling water is piped into the radial header pipes through "bobbin" and O-ring connectors, flows through the helical tubing and leaves through the radial header pipes and connectors at the other end. The helium cycle working fluid flows axially across the finned tubing.



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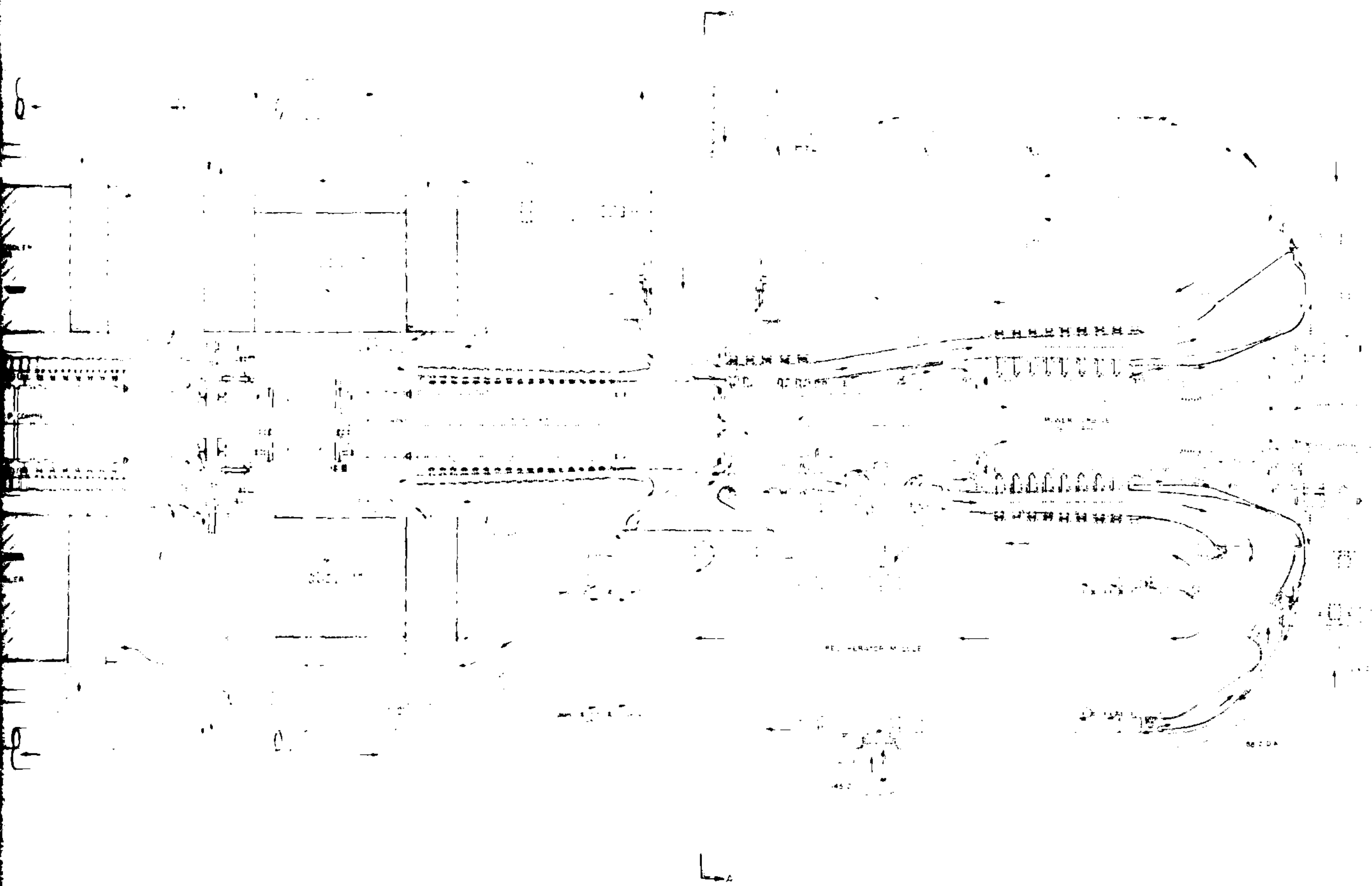


Figure 3-5. Power Plant Layout Drawing 712J486

The intercooler matrix is accommodated within its own cylindrical pressure shell which is terminated at each end in pressurized headers. The headers convey the helium gas from the low pressure compressor outlet into the intercooler and from the intercooler into the high pressure compressor inlet. The intercooler/header/vessel assembly is supported by a conical support member from a flange on the power plant pressure vessel and is sealed to the turbomachinery by means of elastomer O-rings. Complete separation of the higher pressure gas in the intercooler from the low pressure gas entering the precooler is thus achieved while permitting the turbomachinery to be freely withdrawn from the assembly through the O-ring seal interfaces. The concical support member is penetrated by a number of large diameter holes which allow the passage of the low pressure helium from the recuperator to the precooler inlet. The precooler and its headers are integrated into a similar assembly and supported from a flange on the pressure vessel. The coolers have been carefully located in the axial direction to avoid interference with the radial pipes supplying the pressurizing gas to the gas bearings. Pressurizing gas is fed to the bearings through the smaller diameter pipes which are accommodated inside the larger turbine cooling and balance piston supply pipes.

The center frame structure consists of a short cylindrical section of casing having two diaphragm members, one at each end, extending radially inward towards the center. Central holes in the diaphragm members provide radial support for the turbomachinery casing. The two diaphragms are each supported in the axial direction by eight essentially radial webs. Each diaphragm is pierced by seven holes, located midway between the webs. These holes locate the seven recuperator modules. The eighth inter web location accommodates the ducting connecting the assembly to the heat source. A penetration on the exterior cylindrical portion of the center frame structure provides a passageway for the concentric inlet and return ducts to the heat source.

During operation, the center frame structure is pressurized by high pressure compressor outlet gas and serves as the recuperator high pressure inlet plenum. Sealing of the turbomachinery to the center frame is effected by elastomer O rings, carried in grooves machined into the turbomachinery casing, which engage the prepared surfaces at the inner diameter of the center frame

diaphragm members. The gas temperature at this location is approximately 138°C (280°F).

Flanged tubular members are attached to the rear diaphragm of the center frame structure at each of the seven recuperator locations and at the eighth delivery pipe location. The tubular members support a rectangular section toroidal vessel which functions as the recuperator high pressure outlet plenum. The recuperator modules are constructed in the form of thin walled cylinders containing bundles of heat transfer tubing and are inserted into holes in the rear of the toroidal vessel, extending through the vessel into the support tubes and center frame structure. The flow configuration is that of a counterflow heat exchanger with the turbine exhaust helium flowing in the tubes and with the compressor exit helium flowing on the shell side. Relative thermal expansion between a recuperator module and its support members and between the tubing and shell of the recuperator module is accommodated by sliding joints sealed by O-rings at the cold $\simeq$149°C (300°F) end. The modules are axially located and sealed at their hot ends by means of bolted flanges.

The toroidal vessel forming the recuperator high pressure outlet helium is connected to the outer annulus of the concentric duct to the heat source at the eighth delivery pipe location. Two check valves are located at the entrance to the delivery pipe to prevent backflow of gas from the operating unit in the event of the shutdown of one unit of a two unit system. A short inner pipe connects the check valves to the annular duct and is sealed to them by means of piston rings. A piston ring is also used to seal the annular duct to the center frame. The piston rings allow the pipes conveying the hot gas from the check valves to the heat source to expand and contract independently of the relatively cool center frame structure.

The rear section of the pressure vessel supports and axially locates the turbomachinery at the power turbine outlet end. The turbomachinery receives additional support in the radial direction from the center frame structure and from the forward section of the pressure vessel. The rear section of the pressure vessel also supports an electric generator in one of the possible compact closed cycle arrangements.

The turbomachinery is designed to be insertable into or removable from the power conversion assembly as a unit together with the generator and the rear section of the pressure vessel. In a horizontal installation, the pressure vessel rear section serves as a lifting fixture and allows the complete turbomachinery assembly to be suspended at its center of gravity from a crane or transporter device while it is removed axially from the power conversion assembly.

Alternatively, in a vertical installation, the turbomachinery can be lifted vertically from the power conversion assembly, suspended from the generator end. A small diameter hatchway, located in the deck of the ship immediately above the unit could facilitate the rapid removal and replacement of the rotating machinery.

During removal of the turbomachinery from the power plant, the internal flow path is separated by means of sliding joints at the low pressure recuperator inlets. A conical sheet metal vessel confines the relatively warm $\simeq$510°C (950°F) turbine outlet gas to the immediate vicinity of each recuperator inlet. Each conical vessel receives hot gas from the power turbine outlet diffuser through a radial pipe which passes through the turbomachinery rear support structure. The interior surface of the rear section of the pressure vessel and the exterior surfaces of the toroidal recuperator high pressure outlet plenum are exposed to cooler $\simeq$204°C (400°F) gas originating from the power turbine balance piston labyrinth seal.

The removal of the turbomachinery from the power plant requires that the coaxial duct connecting the turbomachinery to the heat source be retracted together with the various auxiliary cycle gas lines to the turbomachinery (used for such purposes as bearing lube and loss of load control).

3.4 CCCBS CHARACTERISTICS

The operating characteristics for the reference CCCBS configuration were developed from the system analyses reported in Section 6.0. These characteristics are summarized in Section 3.4.1.

Due to its compact design and low specific weight, the CCCBS has the potential of being used to meet a wide variety of propulsion needs. Most likely changes in the top level requirements (i.e. power output, plant lifetime, heat rejection, etc.) would be needed due to the specific propulsion application. The effect of variations of the top level requirements on the CCCBS size and state points were analyzed and are described in Section 3.4.2.

3.4.1 REFERENCE CCCBS CONFIGURATION

The reference CCCBS operating characteristics were obtained from the steady-state and transient analyses reported in Section 6.1. The system responses to selected input variations and transient initiators are discussed in detail in that Section. The system statepoints for the reference configuration are stated in Table 3-2.

The primary control system requirements for the CCCBS are to be able to control the plant over the normal operating range of 10 to 100 percent of full power, with a desired rate of change of 10 percent of full power per second. Over this operating range helium inventory control is used in conjunction with control of the heat source outlet temperature. Based on the performance of the CCCBS during the throttle rampdown case, there is almost a one-to-one correspondence between the inventory flow rate and the rate of change of power output. For a helium inventory flow of 10 percent of the full power inventory per second, the output power also decreased at the desired 10 percent per second rate. For the rampup case, a helium inventory flow greater than 10 percent per second would be necessary, due to the helium entering the system at the precooler entrance causing the power turbine outlet pressure to rise at a faster rate than the compressor outlet pressure. This has the effect of reducing the turbine pressure drop and hence the output power. Increasing the inventory flow to about 15 percent per second would allow the plant to increase power at the desired rate.

TABLE 3-2

CCCBS STATE POINTS

	Pressure psia	Temp °R	Rate Lb/Sec	Power MW	Efficiency
COMPRESSOR	454 890	560 764	127.9	34.1	.85
INTERCOOLER	890 883	764 560	127.9	34.1	.98
COMPRESSOR	883 1625	560 741	125.3	29.7	.85
RECUPERATOR	1619 1613	741 1301	120.2	88.9	.84
ENERGY SOURCE	1597 1512	1300 2190	121.5	141.2	--
TURBINE	1500 852	2160 1766	124.0	63.4	.90
POWER TURBINE	852 478	1766 1438	124.0	53.2	.90
RECUPERATOR	472 462	1411 880	127.9	88.9	.84
PRECOOLER	462 455	880 560	127.9	53.4	.982

During normal operation, as helium is bled into or out of the turbomachinery, the output power increases or decreases accordingly. However, at the time when the inventory flow ceases there is a small power increase as the turbomachinery flow equalizes in the turbines and compressors. For the rampup case, this condition accelerates the system sightly up to the steady-state power point. For the throttle rampdown case, this condition leads to a brief period when the output power is greater than that desired. This difference is only about 3% of the full power point, however, and lasts for about 10 seconds. This overpower condition can be greatly reduced through ramping the inventory flow off rather than suddenly shutting the inventory control valve. In actuality this would be the case, since the flow controller would begin shutting off the inventory flow as the desired operating point is reached. By a judicious choice of the inventory controller gains, the desired 10 percent per second power rate of change can be achieved without resulting in an overpower condition near the new desired operating point.

Below a 25 percent throttle position, the output power is varied solely by varying the heat source outlet helium temperature. The change in heat source outlet temperature is 160°C (290°F), varying from 755°C (1390°F) at 7.5 percent output power to 915°C (1680°F) at 25 percent power. For a throttle ramp of 10 percent per second, the heat source outlet temperature would have to change about 100°C per second (180°F per second). The CCCBS power conversion system by itself is well able to follow this transient. This rapid a heat source temperature transient may be difficult to achieve with a fossil fired heat exchanger. With a nuclear heat source, the necessary temperature change could be achieved if the heat capacity of the outlet plug shield could be kept as low as possible and if, for a power rampdown case, there was a rapid decrease in the decay heat power. For the cases analyzed, the turbine inlet temperature lagged the reactor outlet temperature by a significant amount because of the shield heat capacity. For instance, a reactor outlet temperature ramp of 3 percent per second resulted in the turbine inlet temperature changing at about a 1 percent per second rate which resulted in the output power changing at about the same 1 percent per second rate.

As shown in Section 6.1.1, in the normal operating range between 25 and 100 percent output power, the steady-state station temperatures throughout the CCCBS stay relatively constant. During transient operation, however, some of the components are subjected to deviations in the station temperatures away from their steady-state values. As would be the case for any other power-producing machinery the recognition of these deviations are part of the normal design process. The deviations from the steady-state operating points will be dissuccsed separately for the various components comprising the CCCBS.

Turbine

The turbine inlet temperature would change at the same rate as the heat source outlet helium temperature (downstream of the reactor plug shield, for a nuclear heat source). The maximum inlet helium temperature rate of change occurs during a heat source flameout condition (or reactor scram). This would cause a rapid drop in the inlet helium temperature and induce a thermal stress in the first blade row. These thermal transients, however, are of a lower degree and rate than is normally expected and designed for in an air-breathing gas turbine. The blades in these machines have been designed to survive in a much harsher environment and to much larger turbine inlet temperature transients than the CCCBS imposes. A combustor flameout in these units would result in the turbine inlet temperature dropping very rapidly to the value of the compressor outlet temperature due to the lack of any significant heat capacity in the combustor. The CCCBS would have a significant amount of heat capacity in the fossil heat source heat exchanger or the nuclear heat source, depending on which one is used.

The major consideration in the turbine design would be in the choice of materials to withstand the 927°C (1700°F) turbine inlet temperature. As discussed in Section 3.5.2.2, a turbine inlet temperature of 927°C (1700°F) is attainable using presently available materials, primarily IN 100 alloy. The maximum turbine inlet pressure occurs at full power with an ultimate heat sink (sea water) temperature of 30°C (85°F), and is about 12,000 KPa (1740 psia). This would occur for the case where a turbine in one power conversion unit is off-design by a conservatively high 5 percent.

Over the entire operating range and for all the transients analyzed, the turbine speed was always equal to or less than the nominal 18,000 RPM. During periods when the inventory flow was being bled out of the CCCBS, the turbine speed was reduced due to the decrease in the turbine flow in comparison to the compressor flow. This turbine flow decrease caused a reduction in the turbine work and therefore caused the turbocompressor spool to decrease speed. A similar phenomenon occurred during periods when the inventory flow was initially bled into the CCCBS. Once again the compressor flow became greater than the cycle turbine flow, and the unit drops down in speed. As the inventory flow reached the cycle turbine, however, the speed began increasing until the inventory flow was stopped. This effect can be seen in Figure 6-25, where the turbocompressor spool speed became almost constant once the inventory flow was stopped. As a result it would appear that no normally occuring transient would subject the cycle turbine to higher stresses than those that occur at the full power point. It therefore appers that, in order for the unit to survive any normally expected transient, the cycle turbine would not have to be designed to meet any worse conditions than would occur at the normal full power operating point.

Power Turbine

During normal operation the power turbine inlet temperature tends to drop in a similar fashion to the heat source outlet temperature. The inlet temperature is about 705°C (1300°F) at 100 percent output power, and decreases to about 683°C (1260°F) at 25 percent output power. Below 25 percent output power, the inlet temperature is dependent primarily on the sea temperature and on the sea temperature and on the power level. At the 10 percent output power point, the power turbine inlet temperature was about 605°C for a sea temperature at 30°C, and about 550°C for a sea temperature of 2°C. The temperature reduction was due to the lower sea temperature causing a reduction in temperatures and pressures in the low pressure region of the cycle, and allowing for a reduced reactor outlet temperature. Between 25 percent and 10 percent power, the power turbine inlet temperature fell from 689°C to about 604°C, primarily in response to the change in the heat source outlet temperature. This change was independent of the turbine operating mode (constant or variable speed).

The power turbine outlet temperature variation with output power also varied with the sea temperature and mode of operation. For the constant speed mode, the outlet temperature dropped approximately linearly as the output power decreased from 100 percent to 25 percent. This temperature drop varied from 527°C (980°F) to 499°C (930°F) for a 30°C sea temperature, and varied from 488°C (910°F) to 466°C (870°F) at 2°C. For the variable speed mode of operation, the outlet temperature generally increased as the output power was decreased from 100 percent to 25 percent. This was due to the decrease in the power turbine efficiency with speed. For the constant speed operating mode, the turbine efficiency stayed at approximately 90 percent over the whole operating range, while the variable turbine efficiency decreased steadily from 90 percent at full power to about 75 percent at about the 10 percent of full power point. This efficiency decrease raises the power turbine outlet temperature.

All of the previous discussion has been for normal plant operation. During certain malfunction transients it is possible for the power turbine to be subjected to temperatures and pressures above what is expected during normal operation. Particularly critical would be loss of load accidents, where the power turbine could overspeed.

Prior to the discussion of the specifics of these loss of load accidents, it should be recognized that methods (and combinations of methods) were developed to allow the power turbine to survive such an accident. In addition, if there was a loss of load in one power conversion unit only, the plant could continue operating with the failed unit isolated and the other unit continuing to operate at a reduced level (up to 80 percent of its full power point).

On a loss of load case, the rate of increase of the power turbine speed is largely determined by the rotational inertia of the turbine and its connected load. This total inertia would then be dependent on the application of the CCCBS and the assumed point for the loss of load. A conservative design assumption is that the loss of load could occur at the power turbine output shaft coupling, thus resulting in the maximum rate of change of the turbine speed.

There are two separate loss of load cases that the plant could be subjected to. One is a complete loss of load on both power conversion units. This case would require scram of the reactor heat source (or fuel burners shut off, for a fossil heat source), and the rapid activation of the inventory control valves. As the inventory valves decrease the pressure in the CCCBS units, the turbine pressure ratio would be reduced. This, in conjunction with the increasing power turbine speed, would eventually drive the efficiency to zero. This occurs at two distinct times during the transient; once very rapidly in the transient at about 67% overspeed, when primarily the turbine pressure ratio drives the efficiency to zero, and again near the end of the transient after the inventory flow has been shut off due to the inventory bottles being full.

Following the shutoff of the inventory flow, the turbine pressure ratio was increased due to all the compressor outlet flow being sent to the heat source and hence the turbines. This increased the cycle turbine speed due to the efficiency increasing slightly from the zero value it had previously. As a result the power turbine stabilized at a peak overspeed of 80 percent above the 9000 RPM design point. Based on a stress analysis of the power turbine blading, it appeared that the turbine should be limited to less than this value. This resulted in an investigation to determine other possible methods of controlling overspeed. Methods that would reduce the overspeed to a level more developed, and these are discussed in further detail in Section 6.1.2.4. In addition, the control method chosen to control overspeed was influenced by the analysis of the loss of load in one unit only.

The loss of load case in two units would be more unlikely than the loss of load of one unit. Two units operating in parallel would most likely be driving entirely separate loads, and as a result there would be little likelihood of a fault in one unit being propagated to the other unfaulted unit. A more likely event would be a loss of load in one unit, where it would be desirable that the other unit remains on line. For this case the reactor would not be scrammed (or the fuel burner shut off, for a fossil heat source), and the inventory valves would not be activated due to the desirability to keep the remaining unit on line. This would result in peak turbine overspeeds significantly higher than the 80 percent values obtained for the complete loss of load case.

Two of the most practical methods of reducing the maximum overspeed are to design the power turbine to allow the peak efficiency point to occur at a different operating point, or to add a bypass to divert the helium flow around the power turbine. Using a lower operating point for the turbine design value would result in a penalty of a lower plant efficiency being felt when operating at full power. For example, for the complete loss of load case, shifting the turbine design point down to the 95 percent operating point resulted in a peak overspeed of 60 percent versus the 80 percent overspeed obtained with the reference design. This had the penalty of reducing the overall plant efficiency about 1 percent below the reference value of 36.7 percent. While this method of overspeed protection appears practical for a complete loss of load, it would require the turbine to be operating too far away from its design point at full power for efficient operation if this method was used for overspeed protection during a one unit loss of load.

An improved method of overspeed protection would be the use of a combination of bypass valves and turbine block valves. One possible configuration is shown in Figure 3-6. A pneumatically operated valve would be opened upon receiving an overspeed signal and would bypass the normal power turbine flow to the recuperator inlet. In addition, some of the compressor outlet flow would be bypassed to the recuperator inlet also, thus reducing the thermal shock felt by the recuperator module. In addition, a block valve closes at the power turbine exit thus positively cutting off the power turbine helium flow and stopping the overspeed condition.

As described in Section 6.1.3.4, the activation of this bypass valve arrangement would reduce the turbine overspeed down to a manageable value. In addition, as the compressor helium flow is bypassed to the recuperator inlet, the cycle turbine exit and hence the compressor inlet pressures would rise. This would reduce the cycle turbine work and hence the turbocompressor shaft speed would be reduced. Less and less of the helium flow would leave the unit as the turbocompressor shaft decelerates and results in a decrease in the compressor work. Eventually the check valve in the failed unit will operate and isolate the unit from the rest of the system. For a while following this, however, there will still be helium entering the unit until a steady-state pressure level is reached equal to the reactor outlet pressure, at which time all of the helium leaving the heat source will be entering the operating power conversion unit.

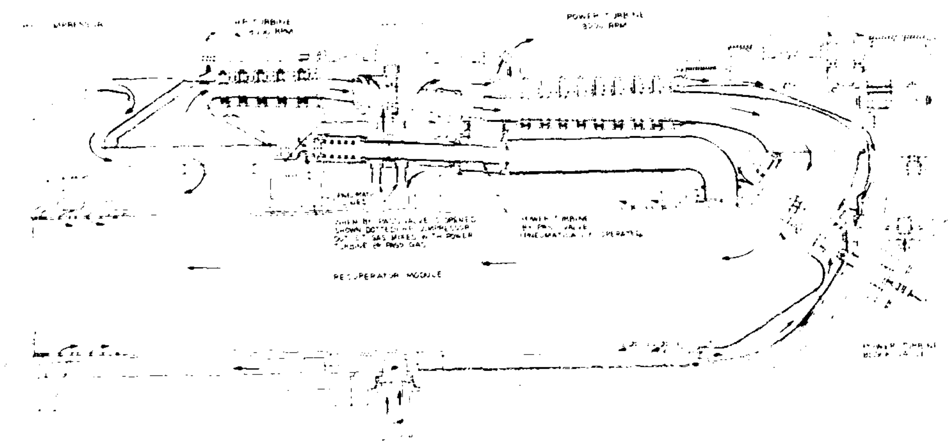


Figure 3-6. Bypass and Block Valve Layout Drawing 102E074

The steady-state operating point would be reached with the failed unit isolated from the rest of the system, the power turbine and turbocompressor in the failed unit coasted down, and the plant continuing to operating on the remaining power conversion unit. However, a disproportionate amount of the total helium inventory would be in the failed unit due to its being totally at the pressure that exists at the cycle turbine inlet. This would allow the remaining unit to operate at only 70 to 80 percent of its full power point.

The loss of load case would also subject the power turbine to the highest thermal stresses. For the complete loss of load case, the peak power turbine inlet temperature is about 733°C (1350°F), and occurs when the inventory flow is stopped due to the inventory bottles being full. This peak temperature is about 30°C (54°F) above the normal design point temperature, and occurs at the 60 percent overspeed condition.

In the loss of load in one unit case, the power turbine could be subjected to temperatures as high as those felt at the gas generator turbine inlet. This would occur after both turbines have coasted down by a significant amount. The reduction in the gas generator turbine work would tend to increase the power turbine inlet temperature in the failed power conversion unit. Even though the turbine flow would be small and the power turbine will have been bypassed, there is a possibility of the power turbine being subjected to temperatures higher than the 733°C (1350°F) value reached in the full loss of load case.

Based on a stress analysis of the power turbine, the present design is able to withstand a 67 percent overspeed condition. There are a number of different methods (and combination of methods) that have demonstrated the possibility of reducing the turbine overspeed to below this limit. For a loss of load in one unit, the plant can successfully be operated with the remaining unit running at between 70 and 80 percent of its full power value.

Recuperator

The thermal stresses that the recuperator is subject to are largely dependent on whether or not inventory control is used. Below 25 percent output power,

inventory control is not used, and the flows stay approximately equal in the low and high pressure sections of the recuperator. This flow equality tends to result in a smoothing out of the thermal transients experienced by the unit. The maximum rates of change of temperature occur at the inlets to the recuperator, and are not governed by their performance characteristics.

When operating above 25 percent output power, inventory control is used as the primary means of controlling output power. During the times when there are rapid changes in the inventory flow, there are large changes in the helium temperatures entering either the low or high pressure side of the recuperator. For example, if there is a sudden influx of helium into the turbomachinery package (as would happen if an inventory control valve between the inventory bottles and the inlet to the precooler were to fail open), there would be a rapid rise in the pressure in the tube side of the recuperator. This would then result in a rise in the pressure at the exit of the power turbine, and would increase the turbine outlet temperature. The fluid temperature entering the recuperator would then rise accordingly. For the cases analyzed, the maximum fluid temperature rise felt at the recuperator inlet was about 125°C (225°F) in about 1 second.

A similar sort of phenomenon would occur for a rapid dump of the turbomachinery helium (as would occur for a fail open malfunction of the inventory control valve between the compressor exit and the inventory bottles). The sudden valve opening would tend to depressure the compressor exit plenum, and hence reduce the temperature of the helium leaving the compressor and entering the shell side of the recuperator. The temperature changes are not as dramatic as the helium influx case, however. The temperature drop is about 50°C (90°F) at the inlet to the recuperator.

These temperature rates would not overly stress the recuperator. The design incorporates a floating end ring at the cold end of the tube bundle. This end ring is sealed to the recuperator shell by the use of O-rings. This allows the cold end of the tube bundle to freely move axially in response to a thermal expansion or contraction of the recuperator tubes.

The worst transient that the recuperator would be subject to would be a loss of load malfunction. For this case the low pressure recuperator inlet area would be subjected to a thermal ramp as high as 220°C per second (400°F per second).

The peak inlet temperature could be as high as the turbine exit temperature, or about 730°C (1350°F). This is assuming that no bypass system is used, which would be needed if one unit operation is desired. Using a bypass system as described in Section 6.1.3.4 would result in lower thermal rates and peak temperatures, since some or all of the relatively cool compressor exit flow would be used to mix with the turbine exit flow. This cooled mixture would then be bypassed around the power turbine and be dumped into the recuperator low pressure inlet plenum.

The recuperator module can handle the high peak temperatures and ramps expected during such an accident. Even without the bypass system, the peak temperature expected (about 730°C, or 1350°F) is still less than 815°C (1500°F) design temperature limit. Also, the use of a floating end ring on the cold side together with O ring seals allows the recuperator tube bundle to readily expand, thereby reducing the stresses that are generated during thermal expansion. It would appear that the recuperators can survive any accident or transient that would not seriously damage any other component in the CCCBS.

Precooler and Intercooler

The precooler and intercooler temperatures are relatively unaffected by transients occurring in other portions of the plant. The high full power effectivenesses of the heat exchangers and the fact that the heat flow is from helium to water tends to smooth out the thermal transients experienced by these components. As long as the water flow is maintained, the region that would be subjected to the worst thermal transient would be the heat exchanger helium inlet. While the intercooler would be relatively isolated from any rapid temperature change, the precooler could be subjected to a sudden 40°C (103°F) drop in the helium inlet temperature if there is a very rapid rise in the inventory flow entering the system. This is due to the relatively cold inventory fluid mixing with the approximately 180°C (355°F) low pressure recuperator exit helium flow. This total temperature rise

would not be felt by the heat exchanger tubes, however. With the coolant flow being water, the tube metal temperature rise would be less than 5°C. Therefore, there would appear to be no helium-side induced transient that would put a large constraint on the design of the precooler or intercooler.

A loss of cooling water accident (either a total loss or an order of magnitude decrease) in either the precooler or intercooler would lead to a rapid rise in the helium temperature exiting the component. In addition, the water temperature in the component would also rapidly rise and eventually boil in the heat exchanger tubes. This could lead to damage of the O-ring seals used to locate the precooler and intercooler around the turbomachinery, and to seal the coolant water header pipes in the unit. Without these seals, leakage of helium from the intercooler to the precooler inlet could occur, and there is a possibility of water mixing with the turbomachinery helium flow.

To improve the reliability of the CCCBS, each of the two power conversion units are provided with their own precooler/intercooler coolant water loops and sea water heat exchanger. In addition, the two water coolant loops are connected together by means of piping and valves. A representative flow diagram for a nuclear powered unit is shown in Figure 3-7. If some sort of a malfunction would occur in one of the water coolant loops, the valves interconnecting the two loops can be opened, and both power conversion loops can continue to operate using the single operating coolant water loop. Due to the fact that now one coolant loop is being used to satisfy the needs of both power conversion units, the maximum power level possible would be at least 50 percent (assuming that the coolant loop was previously operating close to capacity). There would be about a 5°C rise in the cooler exit helium temperature, which would not significantly affect the performance of the plant.

The temperatures described above would occur after a steady-state has been reached in the turbomachinery and one coolant water loop. During the initial transient period when the flow is initially lost in the one loop, there is a chance of boiling occurring in either the precooler or intercooler, or both. This would most likely occur if there is a loss of electrical power to one of the water coolant pumps. This loss of pump power should result in a signal calling for a decrease

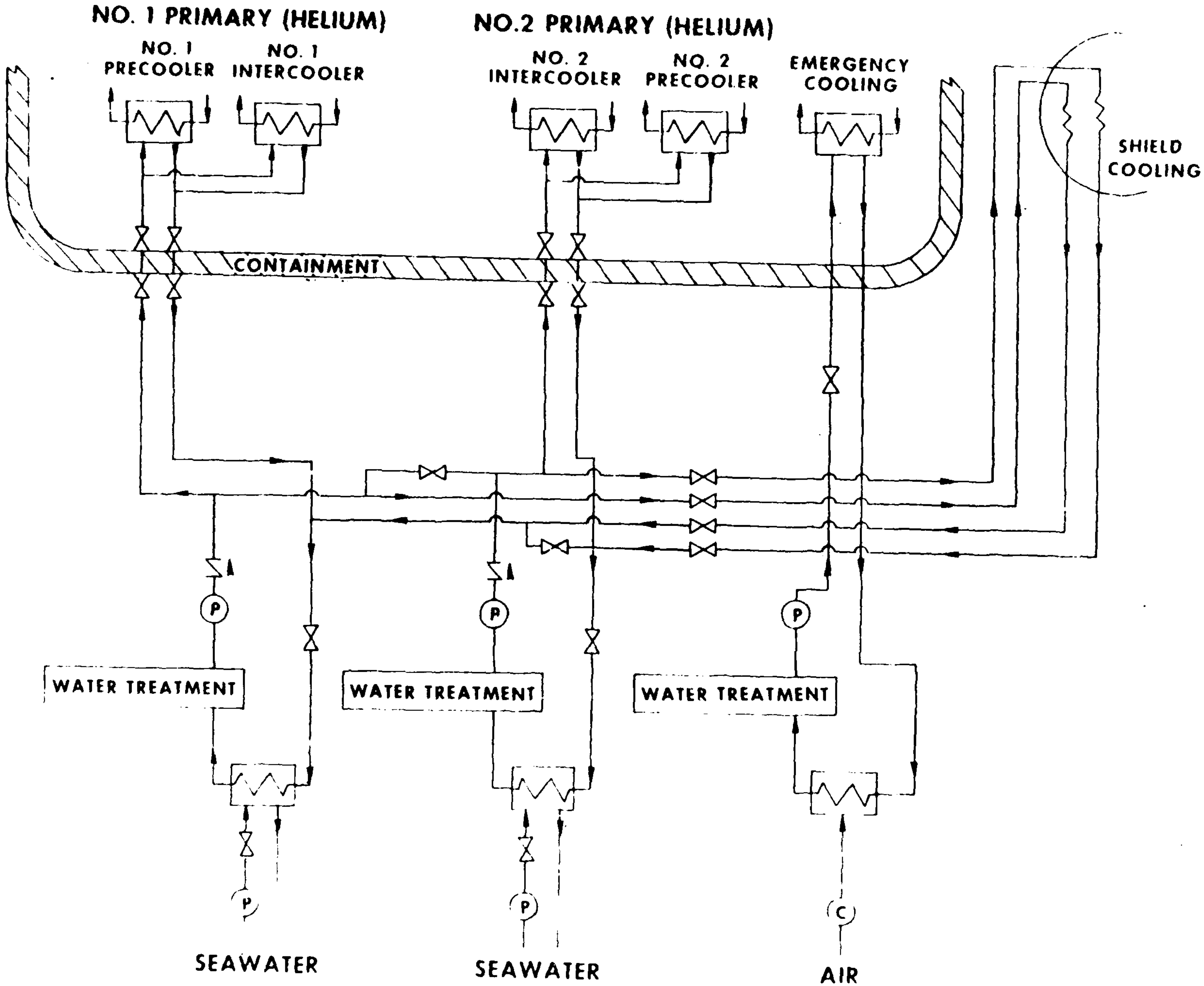


Figure 3-7. Intermediate Cooling Water Loop

in the plant output power and for the opening of the interconnecting valves between the water coolant loops. Before this occurs, the water flow in the heat exchangers would decrease rapidly, and possibly lead to the water boiling in the units. To eliminate the chances of boiling occurring, the pump inertia would have to be large enough so that the water flow can be maintained in the one loop with the failed pump until the other loop can pick up the cooling load. In addition, any boiling that does occur will be of a small magnitude and occur for only about 2 or 3 seconds. This short duration should not lead to any problems with the component integrity.

LP and HP Compressor

A typical compressor performance map is shown in Figure 3-8. This compressor performance characteristic was used in the plant modeling, with the pressure ratio changed to suit the different requirements of the CCCBS.

One of the more serious problems leading to premature compressor failure is a condition known as surging. Under surge conditions, rapid pressure oscillations occur in the compressor, caused by the compressor characteristic (pressure rise vs flow) having a greater slope than the throttle characteristic. Unstable compressor (and hence plant) operation is the result, with the flow rate rapidly oscillating between the stages. To avoid this condition, the compressor must be designed such that it is always operating to the right of the surge line.

During all of the transients analyzed where no bypass system is used, compressor surge was not encountered. The compressor operating points during the transient periods either followed a path roughly parallel to the surge line and passing through the 100 percent design point, or else diverged away from the surge line. In addition, no transient produced turbocompressor shaft speeds higher than the full power design values of 18,000 RPM. As discussed in the section on the cycle turbine operating characteristics, operation during periods of helium inventory flow tends to reduce the turbocompressor shaft speed.

With the inclusion of a bypass system in the turbomachinery to allow for one-unit operation, there is the possibility of surge occurring in the compressor. As the compressor exit and cycle turbine exit flows are bypassed to the recuperator, the

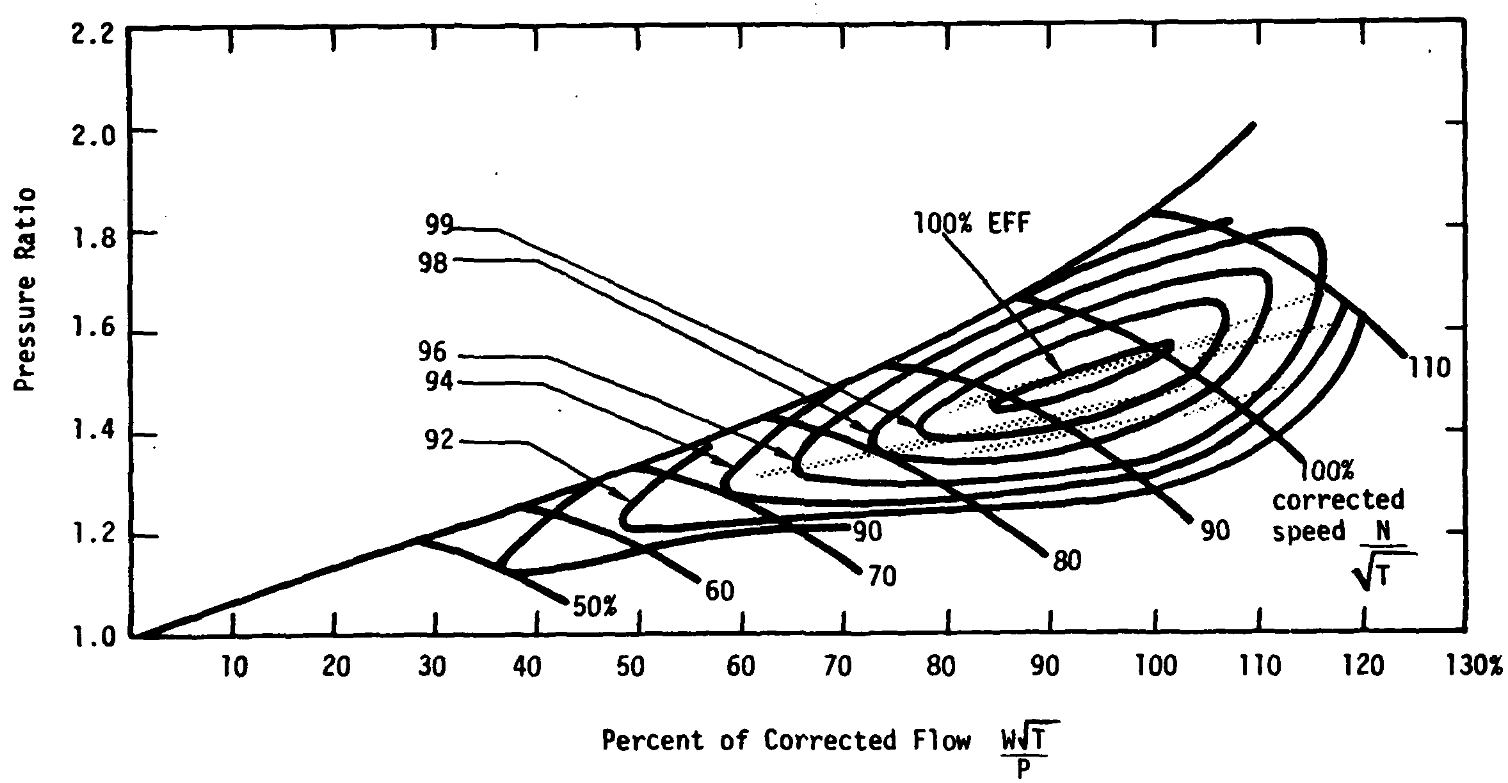


Figure 3-8. Compressor Performance Map

compressor inlet pressure would rapidly rise. In addition, the turbocompressor shaft speed would decrease due to the reduced turbine work. Whether or not surge occurs in the compressor would depend on the rates of change of the compressor speed and flow. A rapid decrease in helium flow coupled with a slow turbocompressor shaft speed decrease would tend to promote surge, while a rapid speed decrease coupled with a slow change in helium flow would result in the operating point diverging away from the surge line. An analysis was performed on a bounding case where both inventory control and a bypass system were used for a complete loss of load case. Here the addition of inventory control tended to rapidly reduce the helium inventory in the turbocompressor package, and hence rapidly decrease the compressor flow. The compressor was not able to decrease speed fast enough, and a marginal compressor surge condition occurred about 2 seconds into the transient. If no inventory control was used, the compressor flows would have been larger, and the chances of surging would have been reduced. In addition, general compressor performance characteristics indicate that the possibility of encountering surge is reduced with increasing percentages of bypass flow. It would therefore appear that compressor surge should not be encountered when operating with a bypass system.

Thrust Bearings

To counteract the axial thrust loads developed by the turbocompressors and power turbines, a combination of gas thrust bearings and balance pistons are used. Using balance pistons tends to reduce the thrust load that must be taken up by the thrust bearings. These bearings were sized to withstand peak thrust loads of 44,500 Newtons (10,000 lb_f) in the turbocompressor shaft, and 97,900 Newtons (22,000 lb_f) in the power turbine shaft.

The thrust load on the turbocompressor shaft is largely a function of the helium inventory flow. During periods when there is no inventory flow, the thrust load varies directly with the output power. The thrust load varies from approximately 6000 Newtons at 10 percent output power to approximately -1000 Newtons at full power (positive thrust defined as being toward the compressor inlet). The use of inventory flow tended to increase the thrust bearing load in a positive direction, and resulted in the thrust load being larger with larger inventory flows. Whether the inventory flow was entering or leaving the turbomachinery

package was immaterial, since both cases result in a positive increase in the thrust load. This was due to the inventory flow causing a reduction in the pressure differential across the balance piston. The peak thrust load was approximately 10,200 Newtons (2300 lb_f), and occurred during an inventory control valve fail open malfunction. Helium flow was simultaneously entering and leaving the system, which resulted in the minimum contribution from the balance piston.

The power turbine shaft thrust load tended to vary with the power produced by the turbine. A peak thrust load of 55,000 Newtons (12,400 lb_f) occurred at the full power operating point. As power was reduced, the thrust load dropped also, reaching a steady-state minimum of 2300 Newtons (500 lb_f) at the 10 percent operating point.

During the times when the inventory system is adding helium to the turbomachinery, the thrust load was reduced due to the reduced pressure differential across the balance piston. If the inventory flow was high enough, the thrust load would become negative (positive thrust being defined as toward the outlet of the power turbine). During normal power rampups at the nominal 10 percent per second, the thrust load stayed positive. However, if one of the inventory control valves that admit helium into the turbomachinery package were to fail open, the large helium flow rate causes the power turbine thrust load to reach a maximum negative value of -12,000 Newtons (2700 lb_f).

In the turbocompressor shaft, the peak thrust load found for any of the cases investigated was about 10,200 Newtons. This is only about one-fourth of the thrust bearing design load of 44,500 Newtons. Likewise, the peak power turbine thrust load of 55,000 Newtons is only about one-half of the design value of 97,900 Newtons. Therefore, the thrust bearings are adequately sized to survive any normal or malfunction condition that does not lead to failure of any other components.

3.4.2 SCALING AND EFFECT OF VARIATIONS IN TOP LEVEL REQUIREMENTS

A set of top level requirements for the CCCBS were defined. These were chosen so that the design and analytical results could be applicable for a wide variety

of propulsion applications. In addition, analyses were accomplished to determine the effect of variations of the top level requirements on the CCCBS operating points and size.

The analyses performed included the following cases:

a) Variation of the 52.2 MW (70,000 SHP) output per Power Conversion Unit.
b) Variation of sea water temperature from 30°C (85°F).
c) Variation of 927°C (1700°F) turbine inlet temperature.
d) Variation of 10.3 MPa (1500 psia) turbine inlet pressure.
e) Variation of heat exchanger (precooler, intercooler, and recuperator) effectiveness.

In the following analyses the effect of variations in top level requirements is shown on the CCCBS power conversion equipment. However, the power conversion equipment only comprises 10 to 15 percent of the total weight of the powerplant and fuel. Therefore any type of optimization should include the total powerplant and the effect of variations in top level requirements on the total powerplant could be significantly different than the effect of variations on the CCCBS power conversion system.

3.4.2.1 EFFECT OF VARIATION OF THE 52.2 MW (70,000 SHP) OUTPUT PER POWER CONVERSION UNIT

Studies have indicated that unit power for the various applications can be expected to range from 14,900 to 74,600 KW (20,000 to 100,000 SHP), with the more likely unit power required in the range of 44,740 to 74,600 KW (60,000 to 100,000 SHP). To make the results of this study more generally applicable, the unit power selected for study should be appropriate for the high power range, but also should be reasonably scalable to lower powers. It was determined that a unit power level of 52,200 KW (70,000 SHP) would fulfill these criteria, and would also provide the benefit to this study of direct infusion of the results of prior Westinghouse funded studies.

In order to determine the effect of scaling on the plant size and performance, the plant operating points and component sizes were obtained for unit powers of

26,100 KW (35,000 SHP) and 104,400 KW (140,000 SHP). In these analyses, the overall turbine/compressor design pressure ratios and the heat exchanger effectivenesses were initially held constant. This was done to avoid having the results unduly influenced by different state points from one case to another.

Using the same heat exchanger geometry, the relation between the specified weight and the unit output power is shown as the solid line in Figure 3-9. For the turbomachinery, the general trend is that the power is proportional to the flow area and the square of the linear dimension, while the weight is proportional to the cube of the linear dimension. This is the primary reason for the increase in the specific weight with increasing output power.

While the above relation holds true for the turbomachinery, it is not quite valid for the heat exchangers. Although the flow area varies in proportion to the power, the length would not change substantially as long as similar tube diameters and geometrical spacing are maintained in the modules. At the lower power levels, unless the heat exchanger geometry is substantially altered, the recuperator would have to be wrapped partly around the precooler and intercooler modules, due to the decreased linear space available. Since one of the design requirements is to enclose the turbomachinery in a pressure vessel capable of withstanding the maximum pressure seen in the cycle, the specific weight would be greatly influenced by the heat exchanger packaging and the resulting length and diameter of the power conversion unit.

In order to decrease the length of the recuperator to result in a more compact package for the 26,100 KW plant, the tube geometry can be adjusted without materially affecting the effectiveness. Among the possible geometrical changes available are reductions in the tube diameter and in the spacing between the tubes. These changes have the disadvantage of increasing the pressure drop through the unit, however, and a reduction in the plant efficiency is the result. One configuration that was tried was to reduce the inner diameter of the recuperator tubes from .254 cm (.1 in) to .203 cm (.08 in), and to reduce the tube spacing from .366 cm (.144 in) to .349 cm (.137 in), and to reduce the heat exchanger effective length to 127 cm (50 in) from the original 178 cm (70 in). While the heat exchanger effectiveness is not affected, the pressure drops are increased by about 14 KPa (2 psi)

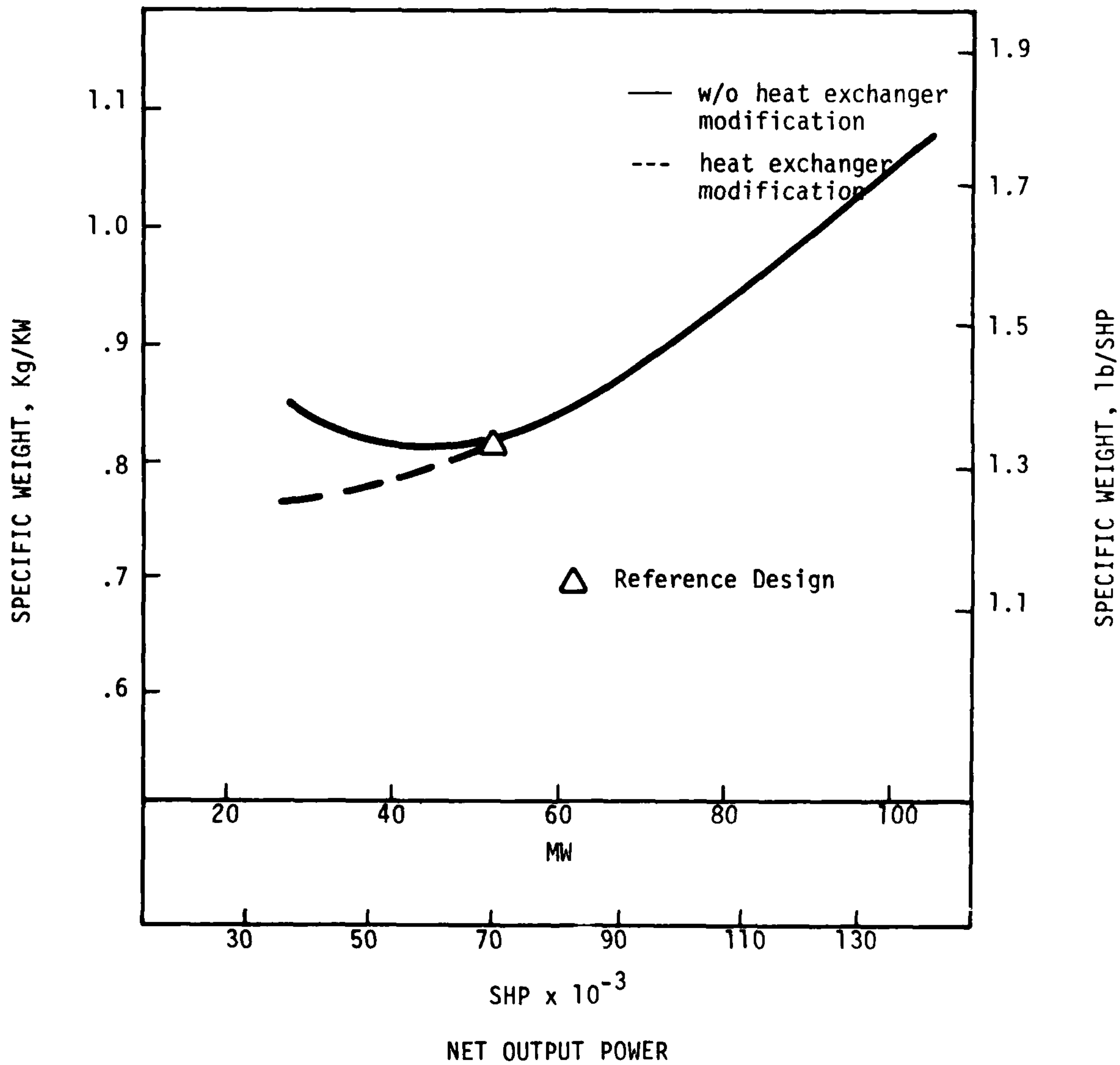


Figure 3-9. Specific Weight vs Output Power

and 69 KPa (10 psi) on the tube and shell side, respectively. This has the tendency to require an increased amount of compressor power, and the overall plant efficiency is reduced by about 2%. Since the precoolers, intercooler, and recuperators can now be wrapped directly around the turbomachinery, the pressure vessel diameter can be reduced, resulting in a decrease in the plant specific weight. The relative benefits are shown in dotted on Figure 3-9.

When scaling above the 52,200 KW reference plant, the length of the power conversion unit is primarily governed by the length of the turbomachinery. As the plant size is increased, there is added wasted space around the heat exchangers due to keeping their length constant. By increasing the heat exchanger length, the space around the turbomachinery can be better utilized, and the outer diameter of the power conversion unit can be reduced. For a 104,400 KW plant, by modifying the heat exchanger design, the specific weight could be reduced to about 0.98 Kg/KW (1.6 lb/SHP), versus the 1.09 Kg/KW (1.78 lb/SHP) value shown in Figure .

Based on the results from this case, the CCCBS can be successfully scaled up or down in power. In order to improve the specific weight of the plant, however, the heat exchangers (precooler, intercooler, and recuperator) would have to be redesigned to optimize the utilization of the available space around the turbomachinery. While a detailed analysis would be necessary at the new power level, it appears possible that the heat exchangers can be designed to best fit into the available space by making relatively simple geometrical modifications. In addition, even without optimizing the heat exchanger designs, the specific weight can be maintained below the 1.22 Kg/KW (2 lb/SHP) requirement for any plant size between 26,100 KW and 104,400 KW.

3.4.2.2 VARIATION OF SEA WATER TEMPERATURE FROM 30°C (85°F)

In the CCCBS analysis, a conservatively high sea water temperature of 30°C (85°F) was chosen as the ultimate heat sink. This would allow the plant to be sized to produce 52,200 KW (70,000 SHP) per unit under any weather conditions that the CCCBS could conceivable to subjected to.

If a lower sea water temperature is used for the design value, the plant would have to be derated when operating at higher sea temperatures. As an example, designing the plant to deliver 52,200 KW at a 2°C (35°C) sea water temperature would enable it to operate at only 85% of the full power level at a sea water temperature of 30°C (85°F). Correspondingly smaller changes in the power level would be noted for smaller differences between the design and the actual sea temperature. The reasons for the design input power reduction at the higher sea temperatures are due to lower cycle efficiency.

If the lower power level at the higher sea temperature is acceptable for the application of the CCCBS, then the plant specific weight can be reduced by designing for a lower sea temperature. For the 52,200 KW plant, reducing the design sea temperature from 30°C to 2°C reduces the specific weight from about 0.817 Kg/KW (1.34 lb/SHP) to about 0.711 Kg/KW (1.17 lb/SHP).

3.4.2.3 VARIATION OF TURBINE INLET TEMPERATURE FROM 927°C (1700°F)

From a plant efficiency and CCCBS specific weight standpoint, it would be desirable to use as high a turbine inlet temperature as possible. However, the use of the higher turbine inlet temperatures makes the attainment of the 10,000 EFPH plant lifetime more difficult. Also, the use of the higher turbine inlet temperatures requires more exotic materials for the piping to and from the heat source, for the cycle turbine inlet plenum, and in the first stage blading of the cycle turbine. Based on these considerations, the use of 927°C (1700°F) turbine inlet temperature was judged to push, but not exceed, the state-of-the-art.

An analysis was performed on the variation of the plant efficiency and CCCBS specific weight with turbine inlet temperature. For these cases the turbo-machinery pressure ratios (with increasing turbine inlet temperatures, the turbine pressure ratio should also be increased to fully utilize the increased temperatures), heat exchanger effectivenesses, and the sea temperature were held constant. The turbine inlet temperature was varied over the range 871°C (1600°F) to 983°C (1800°F).

The plant efficiency variation with turbine inlet temperature can be seen in Figure 3-10. The efficiency varied approximately linearly with turbine inlet temperature, increasing about 1% for each 28°C (50°F) rise in turbine inlet temperature. This efficiency increase was due primarily to the rise in the overall cycle temperature ratio, defined as the highest temperature over the lowest temperature seen in the cycle (temperature entering cycle turbine/temperature entering compressor).

The specific weight variation is shown in Figure 3-11. These results assume that the materials used in the construction of the power conversion units would not drastically affect the weight. As shown in Figure 3-11, the CCCBS specific weight would decrease as the turbine inlet temperature is raised. This is primarily due to the decrease in the helium flow that is possible with the increase turbine inlet temperature.

3.4.2.4 VARIATION OF TURBINE INLET PRESSURE FROM 10.3 MPa (1500 psia)

The use of a 10.3 MPa (1500 psia) turbine inlet pressure was established early in the design phase as being reasonable level during the development of the CCCBS. During the second year activities a number of problems were recognized in using a 10.3 MPa inlet pressure. These were primarily that the high pressures would result in large blade gas loadings and bending stresses, and in large thrust bearing loads. As described in Section 3.5.2.2, further analyses of the cycle showed that the blade bending stresses were not a problem, due to the limited stage pressure ratio that is possible with helium. Also, the thrust bearing load problem could be alleviated by using the full compressor pressure ratio across a balance piston to counteract the generated thrust load.

The main effect of using higher turbine inlet pressures would be a decrease in the helium volumetrics flow as a result of the increased pressure level. This would allow the use of smaller diameter piping between components, and lower turbomachinery flow area. The increased pressure level also reduces pressure losses in the heat exchangers (produces sight increase in cycle efficiency). Also, the pressure vessel thickness would have to be increased to compensate for the higher peak pressure differentials felt through the vessel, and the sealing of the heat exchangers might have to be changed to allow for the higher pressure differences between the tube and shell side flows.

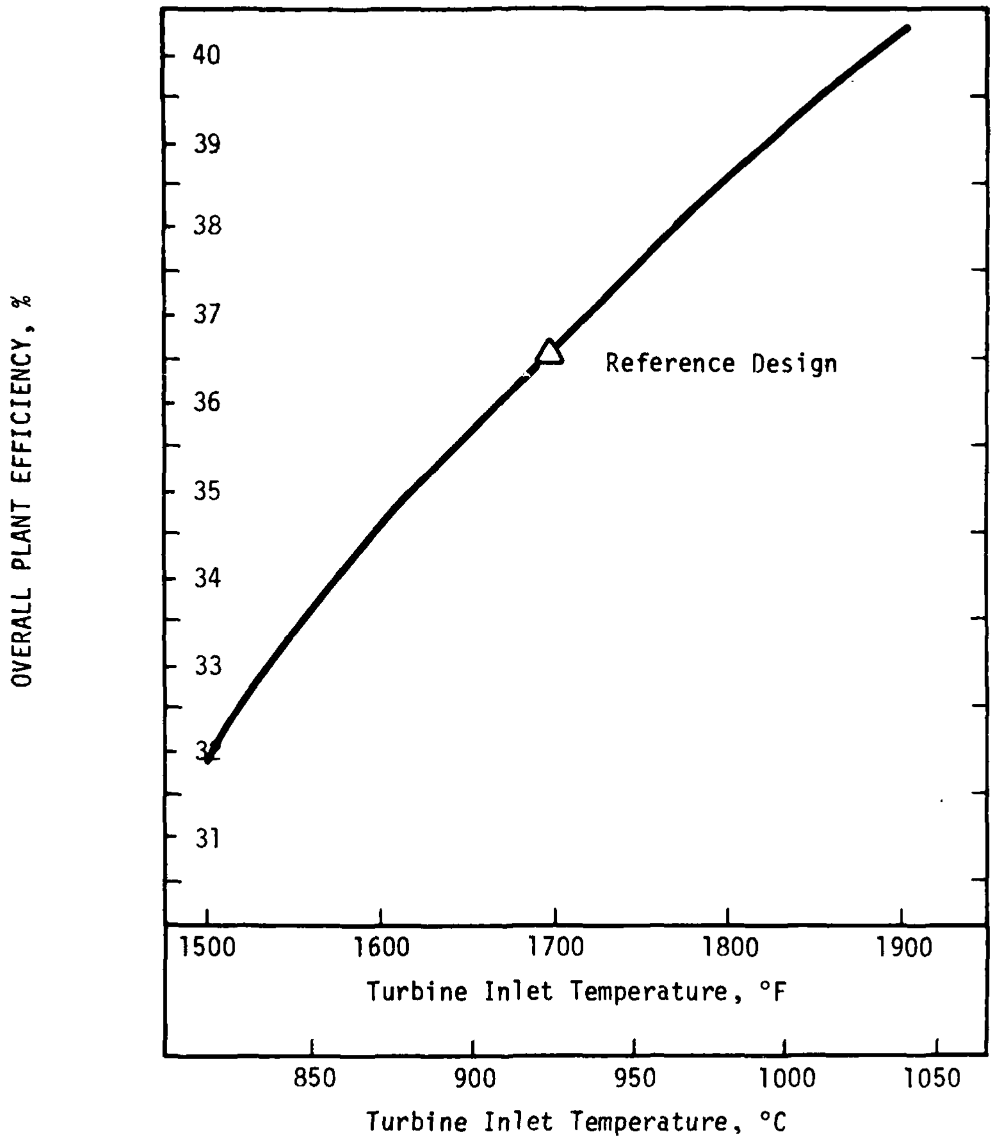
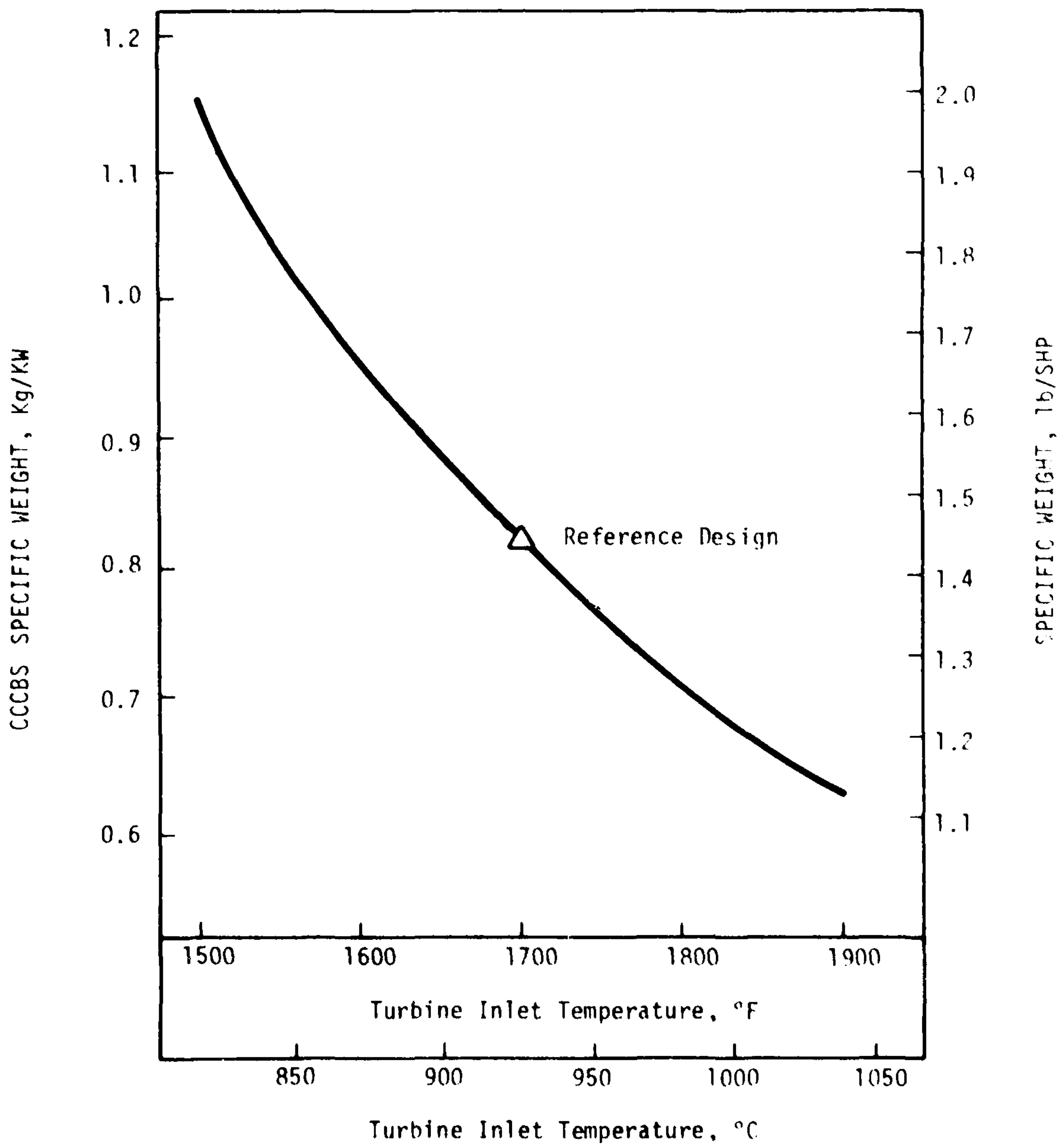


Figure 3-10. Plant Efficiency vs Turbine Inlet Temperature



:1 Specific Weight vs Turbine Inlet Temperature

The relation between the CCCBS specific weight and the turbine inlet pressure is shown in Figure 3-12. Increasing the turbine inlet pressure resulted in a decrease in the specific weight. However, there is a decreasing degree of weight reduction as the inlet pressure is increased.

The analysis presented above is somewhat idealized. Primarily, the heat exchangers were not sized to optimize the specific weight at each new operating point. This tended to result in an increasing amount of wasted space, especially at the higher pressure ratios, due to the proportionately longer turbomachinery. By optimizing the heat exchanger packaging, small improvements can be made in the specific weights at the higher turbine inlet pressures. However, it does not appear possible that a specific weight much below about 0.786 Kg/KW (1.3 lb/sHP) can be achieved at an inlet pressure of 11 MPa (1595 psia). In addition, no account has been taken of any heavier support structures that might be needed internal to the pressure vessel to withstand the higher pressure ratios. This additional weight would tend to increase the plant specific weight slightly.

3.4.2.5 VARIATION OF HEAT EXCHANGER EFFECTIVENESS

During the initial evaluations of the CCCBS, heat exchanger effectiveness of 98 percent in the precoolers and intercoolers, and 80 percent in the recuperators was established as being reasonable. The design phase has borne this out, and effectiveness of 98 percent and 84 percent in the coolers and recuperator can be obtained with the present CCCBS configuration and available technology, and allow the plant to meet the specific weight goal of 1.22 Kg/KW (2 lb/SHP).

The plant efficiency can be increased by increasing the effectiveness of the heat exchangers. However, this is at the expense of added size and weight, and for a compact arrangement an increase in the component pressure drop. Also, each incremental increase in the heat transfer area has a decreasing effect on the component effectiveness as the effectiveness approaches a maximum 100 percent. Eventually, a point will be reached where an increase in the heat transfer area will result in an increase in the overall specific weight, since the added effectiveness is not enough to sufficiently reduce the turbomachinery size and counteract the increased pressure vessel size.

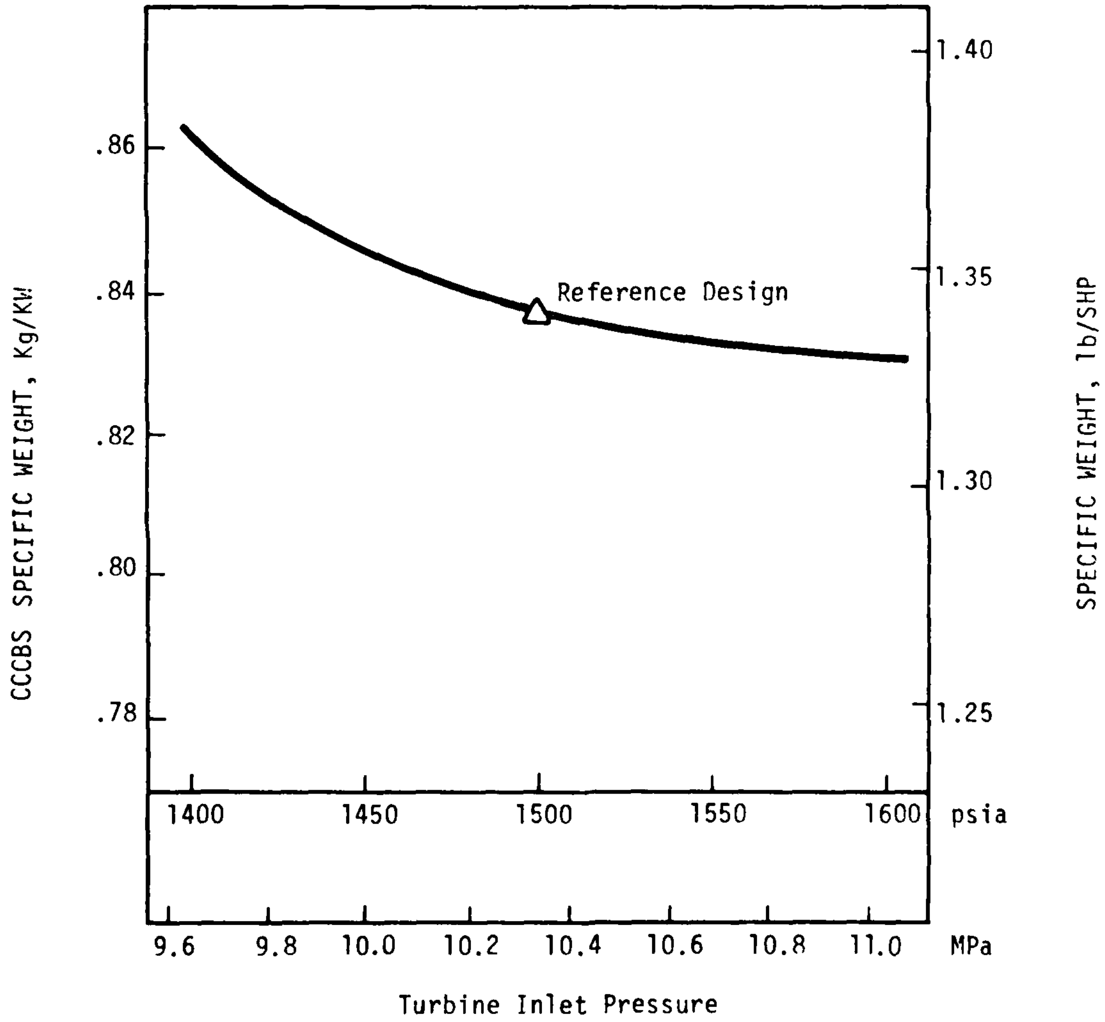


Figure 3-12. Specific Weight vs. Turbine Inlet Pressure

An analysis was performed on the effect of the precooler and recuperator effectiveness on the plant specific weight and overall efficiency. These analyses were limited to changing the total heat transfer area by varying the number of tubes. For the precooler, only decreases in the heat transfer area were investigated, since the precooler effectiveness is already about 98% for the present design. Large increases in the heat transfer area would not appreciably increase the precooler effectiveness or the plant efficiency, and would increase the plant specific weight due to the increased size of the cooler and the resulting pressure vessel.

Analyses were not done on the intercooler, since changes in its heat transfer area would result in similar results to those obtained for the precooler. These coolers are used primarily to reduce the compression power, and similar changes in each component would result in similar affects on the plant.

Figure 3-13 shows the effect of changes in the precooler heat transfer area on the overall plant efficiency. As shown, decreasing the heat transfer area by 40% results in the overall plant efficiency decreasing from about 37% to about 36%. This is the result of the precooler effectiveness decreasing from about 98% to about 91%. The relatively low decrease was caused by the number of transfer units (NTU) being up around 5, and the mass capacity ratio

$$\frac{(w \cdot Cp)_{He}}{(w \cdot Cp)_{water}}$$

being less than 0.28 for the nominal design. A reduction of about 50% in the heat transfer area would reduce the NTU to about 2.5, which does not result in a major change in the effectiveness due to the low mass capacity ratio.

The variation of the plant specific weight with the precooler heat transfer area is shown in Figure 3-14. As the precooler heat transfer area was decreased, the specific weight increased slightly. This is due to the additional helium flow needed to counteract the decrease in the precooler effectiveness, which tended to increase the compressor inlet temperature and therefore the compressor power. The added helium flow increases the size of the turbomachinery and recuperator, which causes an increase in the pressure vessel size. While

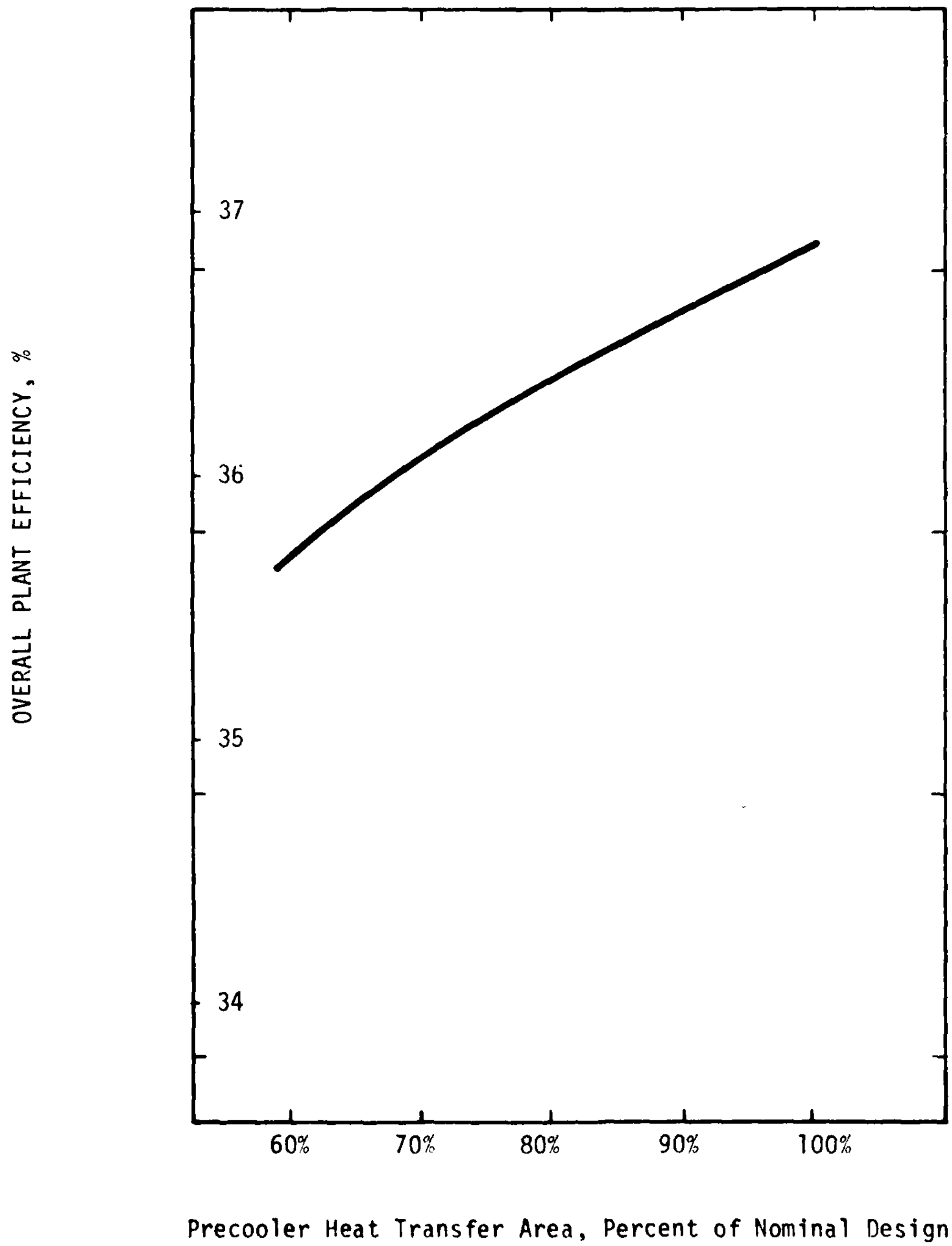


Figure 3-13. Plant Efficiency vs. Precooler Heat Transfer Area

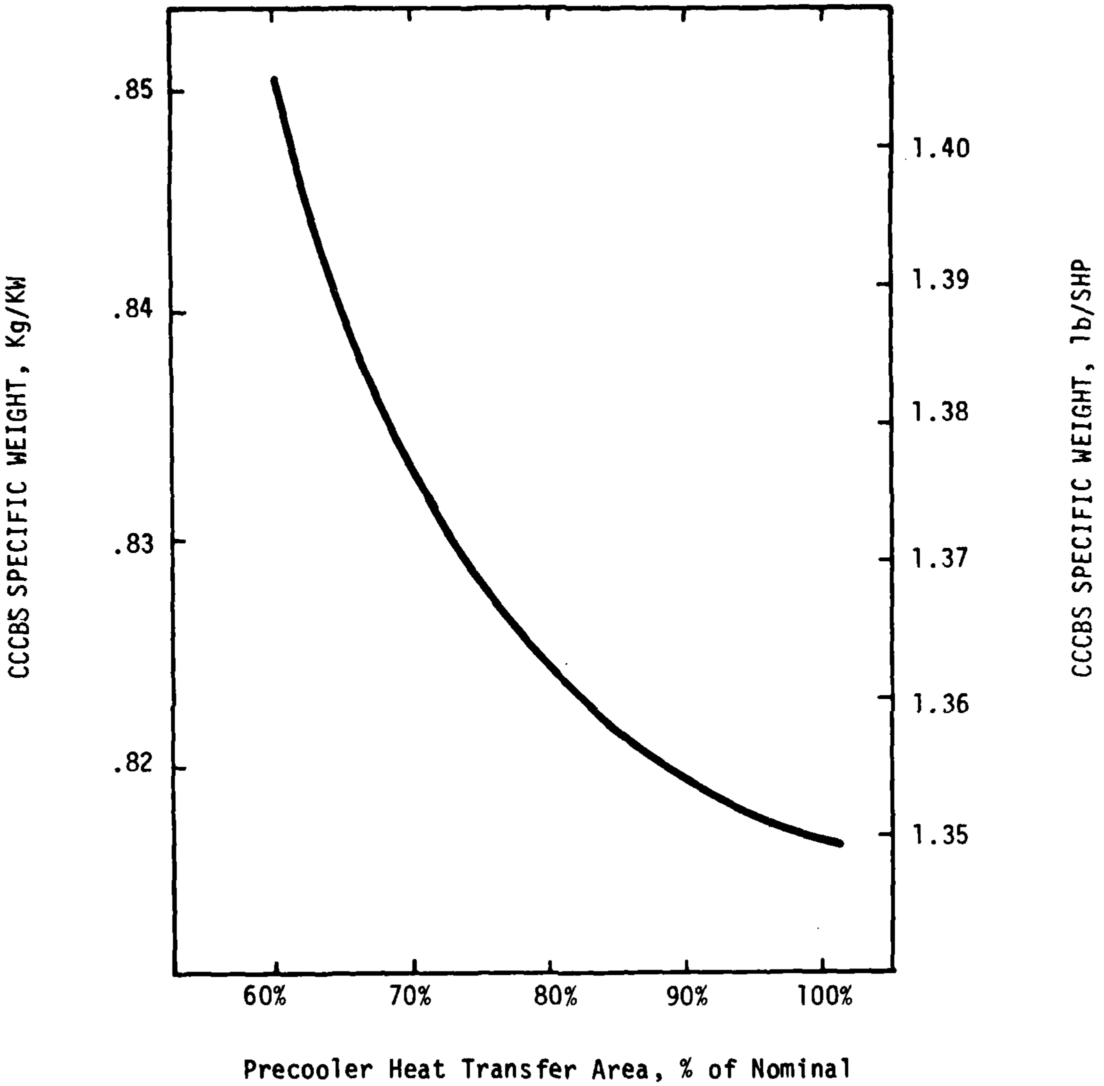


Figure 3-14. Specific Weight vs. Precooler Heat Transfer Area

the decreased precooler size allows the pressure vessel diameter to be reduced in the region surrounding the precooler, it is not sufficient to counterbalance the increased size of the rest of the plant.

The plant efficiency and specific weight variations with changes in the recuperator heat transfer area are shown in Figures 3-15 and 3-16. The heat transfer area was changed by varying the flow areas on both the tube and shell sides. As the recuperator heat transfer area increased, the plant efficiency increased due to the reduced temperature rise needed in the heat source. The rate of efficiency increase tended to reduce with the larger recuperators, due to the heat exchanger effectiveness approaching 100% and being less affected by heat transfer area increases.

The CCCBS specific weight tended to minimize at a heat transfer area about 15% below the nominal design value. Away from this point, the specific weight increased. For the smaller recuperator, the decreased flow area caused the recuperator pressure drop to increase. This resulted in an increasing amount of the total work to be produced in the compressor, and increased the helium flowrates and the turbomachinery size. This increased the size of the CCCBS more than the smaller recuperator was able to reduce it, thus increasing the specific weight. For recuperators larger than nominal, the added heat transfer area and resulting flow area was not sufficient to reduce the pressure drop sufficiently to offset the added weight of the heat exchanger and surrounding pressure vessel. This resulted in the specific weight increasing because of the larger recuperator and surrounding pressure vessel.

The function of the recuperator is primarily to reduce the amount of heat that must be supplied by the heat source. Increasing the recuperator size results in a decreased amount of heat that has to be supplied by an external heat source, as shown by the efficiency curve of Figure 3-15. In evaluating the benefits of using a given recuperator size, the size of the heat source must be taken into account. Depending on the weight-to-power relationship for the heat source, it is possible the overall plant weight and volume could be minimized by using a recuperator size other that the reference value shown in Figure 3-16.

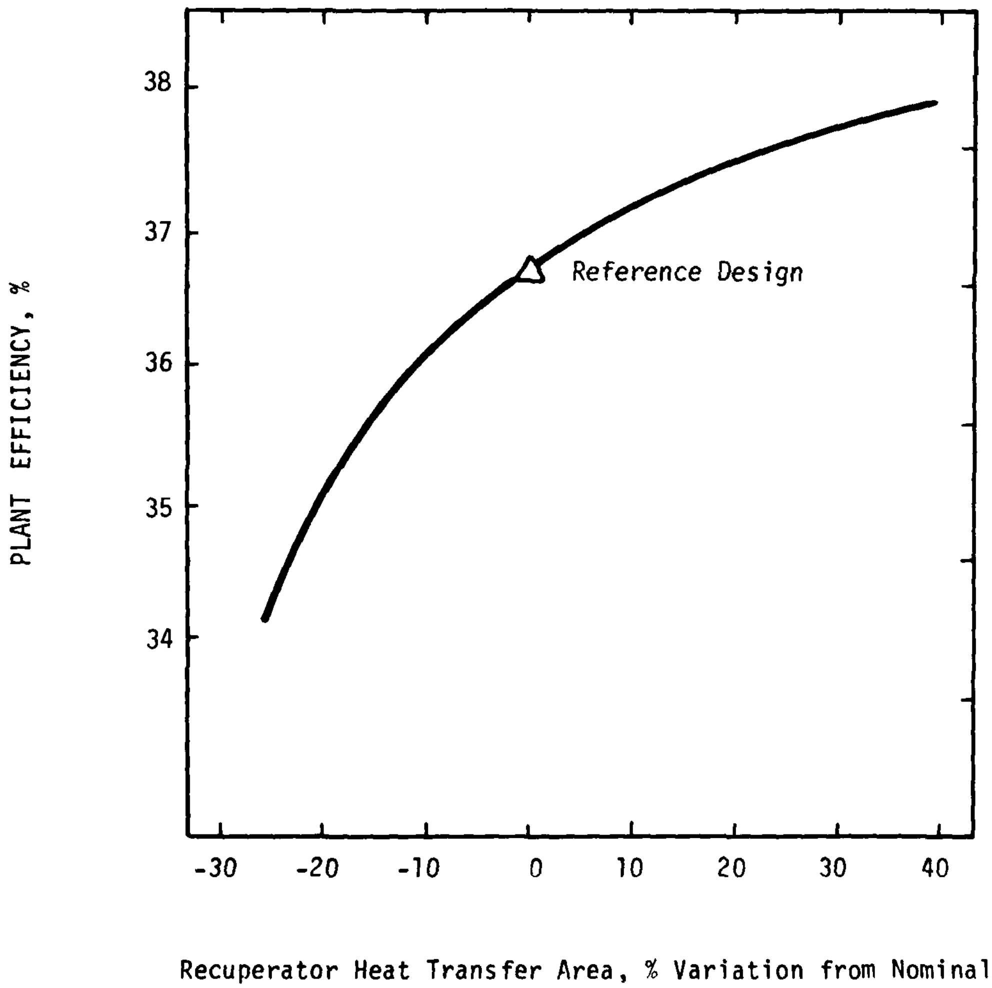


Figure 3-15. Plant Efficiency vs. Recuperator Heat Transfer Area

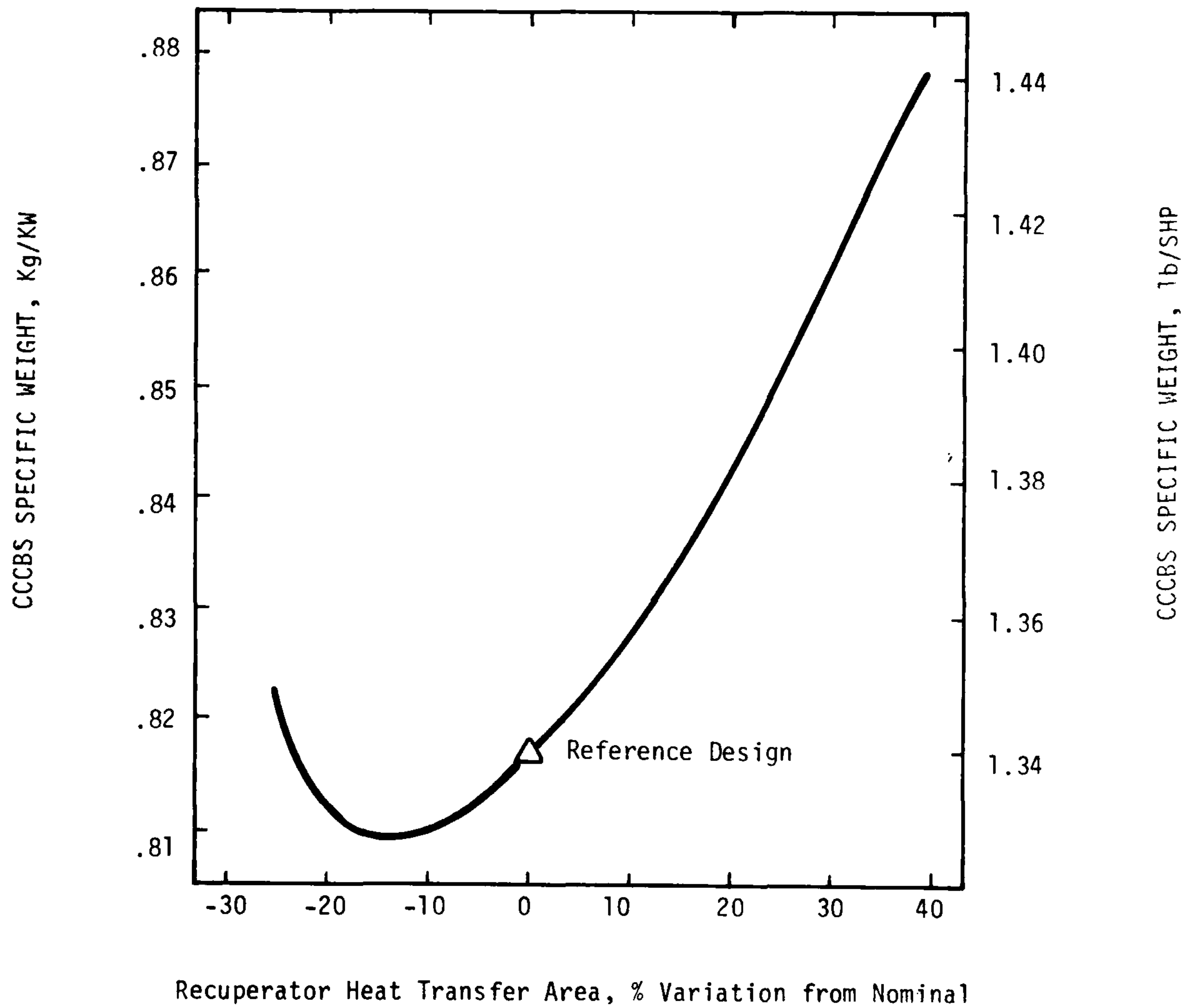


Figure 3-16. Specific Weight vs. Recuperator Heat Transfer Area

3.5 FEASIBILITY EVALUATIONS

3.5.1 OVERALL

Feasibility must be evaluated in the context of the overall program objective and the requirements. Other considerations, not explicitely stated as requirements, but bearing on the utility and general applicability of the systems must also be considered.

The overall objective of the program was to conduct the analytical study and experimental research required to evaluate and to demonstrate feasibility of a closed cycle Brayton power conversion system for a low volume, light weight marine propulsion plant. As will be shown in the following sections the feasibility of the CCCBS concept was proven. The successful demonstration of the plant feasibility therefore allows the development of the CCCBS to proceed, with all of its advantages.

3.5.1.1 ADVANTAGES OF CCCBS

1. Compactness

   The integrated assembly results in a very compact system. The 52.2 MW (70,000 HP) third definition power conversion system has an overall length of 5.5 m (18 ft) and a diameter of approximately 2.3 m (7.7 ft). The resulting power density is 2.28 MW/m^3 (85 HP/ft^3).

2. Light Weight

   The total weight of the 52.2 MW (70,000 HP) power conversion system, with the extra-thick pressure vessel providing two-unit operational capability, was 42,616 Kg (93,755 lb) resulting in a specific weight of 0.82 Kg/KW (1.34 lb/SHP). The weight is reduced to 34,138 Kg (75,103 lb) and, the specific weight to 0.65 Kg/KW (1.07 lb/SHP) if the pressure vessel thickness is reduced to provide only single unit operational capability.

3. High Efficiency Over a Wide Range of Power

   The CCCBS third definition concept has a high overall thermal efficiency ($\simeq$38 percent) which is maintained essentially constant from 25 percent to rated power, in spite of the limitation of the maximum turbine inlet temperature to 927°C (1700°F) to allow the use of uncooled superalloy turbine blades.

4. Adaptable to a Wide Range of Power Outputs and Applications

The CCCBS, as defined in the study, is intended to meet the requirements of high speed ships such as the 200 ton SES. The integrated assembly configuration was adopted to achieve the compactness necessary to minimize the weight of the shielded containment in a nuclear installation. Other CCCBS configurations are also feasible and practical for SES and other applications. These configurations include direct shaft power output, arrangement of system components to minimize height and non-integrated assembly configurations to improve access for maintenance in non-nuclear installations. The study results have provided performance, weight and dimensional data for a 52.2 MW (70,000 HP) powerplant. The studies have also provided a basis for scaling the component designs over the range of approximately 10 MW (13,400 HP) to 150 MW (201,000 HP). In many cases, alternative component designs have been identified which can be considered in trade studies to meet specific user requirements. Thus the CCCBS is adaptable to a wide variety of applications including submarines, displacement vessels, hydrofoils and air cushion vehicles.

5. Not Susceptible to Damage from Salt Spray

The CCCBS has the advantage, for naval application, of being completely unaffected by the external environmental conditions. Salt spray cannot enter the closed system and the turbomachinery and heat exchanger components are thus protected from its corrosive and performance degrading effects. This is particularly true in the integrated assembly design where the external vessel affords substantial protection and could conceivably allow operation in a flooded engine compartment (assuming that the rest of the plant is likewise able to operate under these conditions).

6. Compatibility with Navy Requirements

The CCCBS powerplant also posses many other advantages which enhance its compatibility with Navy requirements. These have been discussed at some length in the feasibility evaluation and include ruggedness, quietness, safety, tolerance to component malfunctions and compatibility with available technology.

7. Future Potential

As a result of the advantages discussed above, the CCCBS is believed to have outstanding potential as an advanced powerplant of the future. Its compatibility with available heat source technology, in the form of either high temperature gas cooled reactors or lower temperature fossil fuel heated heat exchangers, allows the development of this potential to be started immediately. The successful development of a CCCBS power conversion system of the type evaluated in the study would make available an advanced powerplant capability, compatible with a new generation of high speed ships. The resulting CCCBS technology would be applicable

in both fossil and nuclear propulsion plants over a wide range of power outputs and could achieve standardization of propulsion plant for a wide variety of installations.

Since power output is controllable by helium inventory adjustment (pressure level) without affecting other cycle parameters such as temperatures, pressure ratios, etc., a large range of power outputs can be provided for by a single power conversion system hardware development program. Growth potential is available by increasing pressure level. De-rated plant for lower power applications can be made weight-effective by merely reducing the thickness of the pressure vessel components. Inventory control and storage provisions would be provided to suit the power level and range of efficient operation required in a specific application. The resulting ability to use a single power conversion hardware development in a wide variety of high and lower power applications would be cost effective from the standpoint of maintenance, spare parts inventory, etc., in operation as well as in development and manufacture. The future potential of the CCCBS would therefore appear to be very high in view of its performance advantages and its amenability to cost reduction through standardization.

8. Technology Level

Based upon this study, the CCCBS design is well within the present materials and technical state-of-the-art. No technological breakthroughs are necessary for development of any components. The plant development program could commence without a large scale research program being needed.

3.5.2 FEASIBILITY WITH RESPECT TO DEFINED REQUIREMENTS

The top level requirements on the CCCBS were selected to be representative of the range of applications which could reasonably be expected, but sufficiently stringent to test the realism of a closed Brayton system for potential naval propulsion applications. Also, the additional constraint that the requirements be consistent with existing technology or technology being developed was imposed. These requirements have been described before in Section 3.2.1. Only the feasibility with respect to these requirements will be discussed. Several of the specified requirements in Section 3.2.1 impose constraints or limitations which tend to make the other requirements more difficult to satisfy. In this respect the requirements are interdependent. As a consequence of this interdependence it is appropriate, in some cases, to evaluate the feasibility of meeting a particular requirement in relation to the other specified requirements.

As an example of this interdependence, the 30°C (85°F) seawater rejection requirement results in a larger helium flow rate than would be the case for a lower seawater temperature, increasing the component sizes and weight and making the achievement of the 1.22 Kg/KW (2 lb/SHP) specific weight requirement more difficult. Similarly the second level requirement of 927°C (1700°F) turbine inlet temperature makes the top level requirement for 10,000 EFPH (40,000 operating hours) more difficult to achieve than would be the case using a lower turbine inlet temperature. Conversely the 1.22 Kg/KW (2 lb/SHP) specific weight requirement is made easier of attainment by the use of the 927°C (1700°F) turbine inlet temperature than would be the case using a lower temperature.

Recognizing this interdependence, the feasibility of the system is discussed below in relation to the requirements as they are organized in Section 3.2.1 only where the interdependence of the requirements does not make the approach reasonable.

3.5.2.1 TOP LEVEL REQUIREMENTS

1. Reference Application - Surface Effect Ship

   Feasibility of the CCCBS for the Surface Effect Ship application has been clearly demonstrated by the compact, lightweight designs achieved in the study. The designs provide the access to the powerplant necessary for maintenance and accommodate the load variability associated with the application by the use of a free power turbine. Matching electric power and mechanical power transmission systems have both been provided for by alternative 9000 RPM and 3600 RPM power turbine designs. These aspects of feasibility are discussed more fully in the following sections under the particular requirement headings.

   The design features adopted in the study to meet the requirements of the SES application do not exclude other applications of the powerplant. However, a change in the reference application - from the surface effect ship to a completely different application would change the emphasis of several other top level requirements. The power output requirement would almost certainly be substantially reduced from the 52.2 MW (70,000 SHP) value required in the SES application and the specific weight requirement would probably not be any worse than the 1.22 Kg/KW (216/SHP) CCCBS requirement. The other top level requirements (lifetime, shock and heat rejection seawater temperature) would be less likely to be affected substantially. The effect of changes in the power and specific weight requirements on the capabilities of the CCCBS power conversion system are discussed in the two sections immediately following.

2. Unit Power Conversion Assembly Output - 52.2 MW (70,000 SHP)

Feasibility of the CCCBS power conversion system for the required power output is established by the component and system design evaluations performed during the study. Conservative component efficiency and cycle parameters were based on MGCR (Reference 1) component testing experience and other studies. Allowance was made for design compromises made to achieve compactness in the turbomachinery. Allowance was also made for bleed flow rates used for cooling, rotor thrust balancing and bearing requirements. Thus, the CCCBS power conversion unit is conservatively sized to produce the required power under the most adverse conditions forseeable and at 30°C (85°F) seawater conditions. This conservative design approach did not prevent the achievement of a specific weight well within the specified requirement. The resulting cycle thermal efficiency is 37 percent.

In considering the variations in expected capabilities of the CCCBS power conversion system with changes in the output power requirement, the most important influence is the effect of component size on efficiency and specific weight. Capability of the CCCBS in terms of thermal efficiency could be adversely affected by the reduced component size required in a substantially reduced power application. This could result from the relatively large influence of manufacturing tolerances on blading shapes and also, in some cases, from Reynolds number effects. Both of these effects tend to reduce component efficiency in smaller size turbomachines. The reduced component efficiencies, in turn, cause the overall thermal efficiency of the powerplant to be degraded.

Even with the trend of decreasing turbomachinery efficiencies with reductions in the size of the units, there is still the possibility of obtaining improvements in the CCCBS specific weight by reducing the output power requirement. For the turbomachinery, the output power is proportional to the flow area and the square of the linear dimension, while the component weight is proportional to the cube of the linear dimension. Therefore, the turbomachinery weight would vary as the output power to the three-half power (assuming that the blade speed and gas bending stresses are unchanged). This shows that the turbomachinery specific weight would decrease as their size is reduced. However, as the components are further reduced in size, the influences of the reduced turbomachinery efficiencies come into play and tend to drive up the plant specific weight.

In the heat exchanger components, the weight of the unit usually varies directly with the output power, since for a constant heat exchanger length and geometry the flow area would vary directly with power. The specific weight of the heat exchanger components therefore would not vary greatly with changes in the output power.

Generally speaking, therefore, when scaling the CCCBS power conversion system, the specific weight will tend to reduce somewhat with reduction in power level due to the turbomachinery scaling

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influences. However, it must be borne in mind that these trends are somewhat idealized and may be overshadowed by more mundane considerations in a practical design situation. In the case of the integrated assembly arrangement, chosen primarily for its compactness in a nuclear installation, the influence of the external containment vessel weight can be obliterate that of the turbomachinery if, in scaling down the plant, the smaller system incurs a larger percentage of wasted volume inside the containment vessel. This situation can arise due to the relatively smaller length achieved in the turbomachinery compared with the heat exchanger, which makes the heat exchangers difficult to fit economically, together with the turbomachinery, inside an enveloping pressure vessel. Consideration must therefore be given to modifying the heat exchanger design basis to achieve component geometry which is more compatible with the integrated assembly. In this content, the use of finer matrix geometry may be appropriate in designing for reduced power output.

Analyses done up to the present time (described in Section 3.4.2.1) indicate that the specific weight of the CCCBS power conversion unit tends to decrease with a decreasing plant output power design requirment. This is mainly due to the reduced turbomachinery and pressure vessel sizes. Below a plant size of about 26,000 KW (35,000 SHP), however, the specific weight would start increasing as the decreased turbomachinery efficiency comes into play.

3. <u>Specific Weight of Power Conversion System - 1.22 Kg/KW (2 lb/SHP)</u>

Feasibility of the CCCBS power conversion system to meet the required specific weight of 1.22 Kg/KW (2 lb/SHP) has been demonstrated by the results of a detailed weight estimate of the third definition design concept, illustrated in Figure 3-5. The total weight is 42,519 Kg (93,755 lb) resulting in a specific weight of 0.83 Kg/KW (1.34 lb/SHP) which is less than 70 percent of the specified requirement. Moreover, even this low weight already includes the extra pressure vessel weight which results from the required capability to operate in a two unit system. If such capability were not required, the pressure vessel thickness and weight could be substantially reduced, reducing the total weight of the system to 34,060 Kg (75,103 lb) and the specific weight to 0.65 Kg/KW (1.07 lb/SHP).

The version of the CCCBS power conversion system with the 3600 RPM power turbine illustrated in Figure 7-4 was heavier than the 9000 RPM design at 50,233 Kg (110,763 lb) and its specific weight was 0.96 Kg/KW (1.58/SHP). These values would reduce to 40,886 Kg (90,153 lb) and 0.78 Kg/KW (1.29 lb/SHP) respectively in a system designed for single unit operation only. All of these values are well within the requirement of 1.22 Kg/KW (2 lb/SHP) established for the SES application.

In considering the variations in expected capabilities of the CCCBS power conversion system with changes in the specific weight requirement, it is clear that the difficulty of achievement increases as the required specific weight is reduced. At the SES

application power level considered in the study, if the required specific weight were to be specified significantly below the region of 0.61 - 0.91 Kg/KW (1-1.5 lb/SHP), changes would have to be made to the power conversion system design to reduce weight. These changes could take many forms, depending on which of the powerplant capabilities could be compromised.

As an example, weight could be reduced further by departing from the conservative design practices and materials recommended in the ASME pressure vessel code and adopting higher strength materials and less conservative design margins associated with aircraft engine practice. However, the effect on life and the need for development testing, as in aircraft engine programs, would have to be evaluated using this approach. It would appear that a light weight, yet rugged and reliable, plant could be developed using aircraft engine practice by putting increased emphasis on the plant lifetime requirement.

Another step which could be taken to reduce weight, if operating life could be compromised, would be to increase turbine inlet temperature, thus achieving a higher power output per unit mass flow and so enabling the powerplant mass flow for a given power to be reduced. The mass flow reduction, so achieved, would enable the component sizes and weight to be correspondingly reduced. Performance of the powerplant, in terms of thermal efficiency, would be increased in this case, resulting in further savings in fuel weight for a given mission. However, operating life is reduced severely as turbine blade metal temperature increases, each 55°C (100°F) in temperature causing approximately an order of magnitude reduction in turbine life. The potential weight reduction achievable as a result of increased turbine inlet temperature is obviously limited by considerations could be reduced if the turbine blades are cooled. However, turbine development costs can be expected to increase, if cooled turbine blades are adopted.

Weight reductions could also be achieved by permitting higher pressure losses and reduced heat transfer effectiveness in heat exchanger components thus allowing their sizes to be reduced. Obviously, these changes are in the direction of reducing plant thermal efficiency and are less desirable, in this respect, than turbine inlet temperature increases. However, some combination of both approaches might be justifiable to meet a specific combination of life and thermal efficiency requirements.

Weight reductions are also achievable in turbomachinery components by adoption of less conservative design practices allowing higher operating stress levels and, possibly, reducing component efficiency. The effects of these approaches on operating life and development cost must, of course, be considered. As an example, blade chord can be reduced to achieve reduced length and weight, but at the expense of increased gas bending stress which may, in turn, adversely effect blade vibration characteristics and potential operating life. Highly stressed blading which exhibits blade vibration tendencies during development testing may necessitate costly changes to develop the required life capability.

The use of thinner sections in turbomachinery casings and rotor components can also achieve weight reduction. However, the effect of the increased stresses on operating life, reliability and also performance, due to potentially increased casing deflections must be evaluated.

4. Lifetime - 10,000 Equivalent Full Power Hours (40,000 Operating Hours)

The feasibility of the 10,000 EFPH requirement must be considered in relation to the assumed turbine inlet temperature. A turbine inlet temperature of 927°C (1700°F) was established as a second level requirement based on the expected capability of uncooled superalloy turbine blading. A capability of 10,000 hours life at the 927°C (1700°F) condition has been predicted, as discussed in Section 8.3.1, based on the air and helium materials test data for IN100 material. To accommodate the increased life requirement at reduced power level, the turbine inlet temperature was scheduled to reduce linearly to 899°C (1650°F) at 25 percent power. The 28°C (50°F) reduction in temperature results in an increase in blade life from 10,000 hours at full power to 40,000 hours at 25 percent power using a Larson-Miller T (20 + Log t) relation between temperature (T) in °R and life (t) in hours.

The feasibility of the CCCBS in relation to these lifetime requirements is considered to be well established in view of the margin of 21°C (37°F) of excess temperature capability, (equivalent to ~7000 full power hours) estimated in Section 8.3.1, of the IN100 material. This margin should be amply conservative in view of the favorable conditions of exposure presented by the helium environment. Combustion turbines impose much more arduous conditions on turbine blade materials in this respect, due to the corrosive combustion gas environment and the more severe temperature transients experienced at start-up, especially when fuel injection nozzle problems are encountered.

The selection of IN100 material as the basis of this evaluation is conservative in that it is a nickel based superalloy which is widely used in current production gas turbine engines. Superalloy materials with improved capability are under development and will be available in time for the CCCBS development program.

While turbine blade temperature is certainly a prime influence on lifetime capability in gas turbine powerplants, consideration must also be given to other potential influences. Temperature conditions high enough to cause concern for metal creep are seen in the inlet duct from the heat source and in the high pressure turbine entry plenum. However, in these areas the components exposed to the highest temperature conditions are essentially unstressed and function as liners to protect the pressure carrying members from the high temperature turbine inlet gas. A potential material for these applications is Hastelloy X, modified in the manner stated below. Hastelloy X has a 10,000 hour rupture strength at 927°C (1700°F) of approximately 14 Mpa (2,000 psi) which is adequate for its essentially unstressed role as a hot liner. This material has been found to be very resistant to cracking and distortion in fossil fuel combustion systems and has good weldibility.

Hastelloy X was adopted by the Japanese Atomic Energy Research Institute (JAERI) as the basic material for high temperature helium service after screening many candidate alloys. Testing of Hastelloy X material at 1000°C (1832°F) in helium with impurities has indicated that improvements can be made by increasing the manganese content of the alloy and minimizing the aluminum, phosphorous and sulphur constituents. Manganese was especially beneficial in improving the adhesion of surface oxide films and improving corrosion resistance under these conditions (Reference 4). The modified material, namely Hastelloy-XR, is considered by JAERI to be a first generation specification for high temperature helium exposure.

In considering the variations in expected capabilities of the CCCBS power conversion system with changes in the lifetime requirement, the effect of the life requirements on the permissible turbine inlet temperature and its effect, in turn, on performance must be reviewed. As discussed in the previous section, based on creep limitations using uncooled blades, an order of magnitude change in life requirement implies, approximately, 55°C (100°F) change in temperature. Thus if the life equipment were to be increased to 100,000 EFPH the turbine inlet temperature would have to be reduced from 927°C (1700°F) to 871°C (1600°F) assuming the use of IN100 material. As a result, the thermal efficiency of the plant would be reduced and the specified weight increased. Similarly, a reduction in the required life would permit an increase in turbine inlet temperature with a resulting increase in thermal efficiency and a reduction in specific weight and volume.

5. Shock - MIL-S-901C (Navy)

A shock analysis in accordance with Reference 5 was performed on the system. This is discussed in more detail in Section 6.2.2. While no problems are anticipated in accommodating the component loads in the integrated powerplant assembly, the resilient foil type gas bearings, recommended at the conclusion of the first year study, have excessively large radial deflection under load, causing blade tip rubbing to occur under shock conditions. This deflection problem has been eliminated in the third definition concept by the substitution of solid geometry pad type, hydrostatic bearings in place of the foil bearings. In this design the approximately 18g effective load produced at the bearings causes metal to metal contact to occur between journal and pad when an effective load of 2-5gs (depending on bearing location) is exceeded. The bearing deflection at this point is limited to the sum of the gas film thickness and the deflection permitted by stops incorporated into the resilient bearing supports. The maximum bearing deflection under shock would be approximately 0.20 mm (0.008 in.) composed of 0.076 mm (0.003 in.) gas film deflection and 0.127 mm (0.005 in.) resilient mount deflection. The gas film thickness and resilient mount travel could be controlled at each bearing to limit the maximum deflection to values that would prevent rotor blade tip rubbing. The feasibility of this approach in the CCCBS cannot be completely evaluated

in the absence of test data. Based upon tests of much smaller gas bearing supported turbomachinery, shock applications of short duration (0.02 mil sec) and several hundred g's can be accepted without failure for at least 100 cycles. The relevance of these tests to the CCCBS high inertia rotors has not yet been demonstrated. As stated in Section 8.1.1, however, it is felt that on acceptable bearing design can be obtained without seriously impacting the CCCBS development program.

6. Heat Rejection - To Seawater at 30°C (85°F)

The 30°C (85°F) seawater requirement is a conservative (high) assumption which has the effect of reducing thermal efficiency and increasing specific weight. However, the feasibility of the closed cycle Brayton power conversion system for a low volume, light weight marine propulsion plant is not unduly compromised by this requirement. The resulting powerplant, in its integrated assembly configuration, is well within the specific weight requirement in spite of any component size and weight penalties incurred by the seawater temperature requirement.

Changes in the seawater temperature requirement would affect the capability of the CCCBS as a result of the change in overall cycle temperature ratio, assuming a fixed turbine inlet temperature. A reduction in seawater temperature would increase thermal efficiency and reduce specific weight. Figure 6-2 shows the effect of seawater temperature on thermal efficiency. The effect of seawater temperature on power and, hence, specific weight is discussed in Section 3.4.2.2.

Thus the 30°C (85°F) seawater condition represents the most stringent requirement and feasibility, having been demonstrated at this condition, is assured for lower seawater temperature conditions.

3.5.2.2 SECOND LEVEL REQUIREMENTS

1. Turbine Inlet Temperature - 927°C (1700°F) and Pressure - 10.3 MPa (1500 psia)

The feasibility of the 927°C (1700°F) turbine inlet temperature requirements must be discussed in relation to the required 10,000 EFPH life. The considerations involved are essentially as discussed in relation to the first requirement of Section 3.5.2.1.

At the time this turbine inlet temperature requirement was established it was judged to push but not exceed the state-of-the-art for uncooled blades using conventional superalloy materials. It was decided not to base the design on the use of refractory materials such as molybdenum which, although apparently ideally suited for use in a helium gas environment, were less well developed and understood than the conventional superalloys. Moreover, to ensure a high degree of conservatism, the design was based on the use of first stage turbine blades made of IN100 alloy, which

is widely used in current open cycle turbine engines. It was recognized that stronger materials, currently under development, would become available in the future, providing growth potential or additional conservatism. The first stage turbine design was refined during the second year program to provide 10,000 hour capability at 927°C (1700°F) turbine inlet temperature. This is discussed more fully in Section 8.3.1. At that condition the first stage blade was estimated to posses a 21°C (37°F) margin of conservation. Because this estimate is based on a single sample of IN100 material tested in 927°F (1700°F) helium, confidence in its precision is obviously limited. However, the additional test data obtained on IN100 in an air environment show similar results and tend to support this estimate.

In order to provide for the increased operating life required at reduced duty profiles the turbine inlet temperature is scheduled to reduce linearly from 927°C (1700°F) at 100 percent power to 899°C (1650°F) at 25 percent power. The reduced temperature of the blade metal at the lower power levels results in appropriately increased life at those conditions to provide 10,000 EFPH over the complete power range. The compromise in efficiency at the lower power levels due to the turbine inlet temperature reduction is small and, in fact, the overall thermal efficiency of the plant is essentially constant (between 36.5 and 37 percent) over the range from 25 to 100 percent of rated power for the constant speed power turbine case.

Thus the feasibility of the closed Brayton cycle power conversion system for 927°F (1700°F) turbine inlet temperature is believed to be well established. Moreover, it is believed that the margin of conservation would be increased in a more optimized turbine design, design refinement having been limited in this program to the degree sufficient only to demonstrate feasibility.

In considering the variations in expected CCCBS capabilities with changes in the turbine inlet temperature requirement, the trends discussed in Section 3.5.2.1, for the top level lifetime requirement are applicable. Assuming the use of uncooled blades, life is reduced by approximately an order of magnitude for each 55°C (100°F) increase in turbine inlet temperature and vice-versa. Increased turbine inlet temperature results in reduced specific weight and volume and increased thermal efficiency, as discussed in Section 3.4.2.3.

Feasibility of the 10.3 MPa (1500 psia) turbine inlet pressure second level requirement is essentially independent of the high temperature metal creep considerations on which feasibility of 10,000 EFPH life at 927°C (1700°F) depends. A major influence of the turbine inlet pressure level selection is felt in the design of the pressure vessel which is required to contain the 11.17 MPa (1619 psia) recuperator inlet pressure associated with the 10.3 MPa (1500 psia) turbine inlet pressure. However, the design of the powerplant is such that the pressure vessel is exposed only to the lower temperature gases in the engine cycle. The highest temperature to which the pressure vessel is exposed

is that of the 216°C (420°F) helium leaving the recuperator. Therefore, the pressure vessel can be made of widely used, well understood, pressure vessel code. An example of such a material is SA 533 which is classified for use at temperatures up to 371°C (700°F). This material has been used in many large pressure vessels [up to 5.5 m (18 ft) diameter and 135 mm (5.3 in.) thick] and is the material used in Westinghouse pressurized water reactor vessels. The diameters and thicknesses required in the CCCBS pressure vessel are on the order of 2.25 m (88 in.) and 51 mm (2 in.) respectively, both well within the state-of-the-art.

An indirect result of the high turbine inlet pressure is a high level of component pressure differential, which was of initial concern to the turbomachinery designers as a possible cause of correspondingly high blade gas loadings and bending stresses. This concern was, however, alleviated by the finding that, in the CCCBS component designs, acceptable gas bending stresses were easily achieved without resort to extreme blade geometry (such as excessively low aspect ratios).. This result is due in part to the thermodynamic properties of helium which limit stage pressure ratios thus distributing gas loads between a relatively large number of stages. However, a similar concern for the large thrust bearing loads expected to result from the selection of the high turbine inlet pressure was found to be quite valid. The unbalanced thrust loads generated by the turbine was found to be quite valid. The unbalanced thrust loads generated by the turbine and compressor components appeared to stretch the state-of-the-art thrust bearing capability and provision had to be made in the design to balance out a substantial portion of the thrust loads by means of balance pistons (dummies). Moreover, initial thrust balancing solutions, using only a portion of the full cycle pressure ratio across the balance piston, were successful only at one design condition and resulted in excessively large variations in thrust bearing load under off-design conditions. Further study of the problem revealed that the use of the full compressor pressure ratio across the balance pistons achieved substantially reduced variations and acceptably low thrust bearing loads over the complete operating power range. Feasibility of the 10.3 MPa (1500 psia) turbine inlet pressure level requirement is therefore established by these evaluations.

Variations in CCCBS capability are less substantial for changes in turbine inlet temperature. Operating life and overall thermal efficiency are essentially unaffected by changes in pressure level. Specific volume and weight are both reduced as cycle pressure is increased. Reduced specific volume is especially beneficial in allowing containment and shielding weight to be minimized in direct cycle nuclear plants. The value of 10.3 MPa (1500 psia) selected for the CCCBS may be unduly conservative in the context of the integrated assembly concept which minimizes the individual component sealing problem. Blade gas bending stresses, using reasonable geometry, have not been found to be severe, and should be maintainable at acceptable levels in a high pressure design without excessive geometry compromises. The pressure vessel wall

thickness of approximately 51 mm (2 in.) could certainly be increased substantially without adverse effects on manufacturability or reliability. Also the selection of a substantially higher pressure should not render thrust balancing impractical, assuming the use of the full compressor pressure ratio for the balance pistons.

2. Integrated Assembly

The integrated assembly requirement must be considered in relation to the overall program objective which is stated as follows:

> "Conduct the analytical study and experimental research required to evaluate and to demonstrate feasibility of a closed cycle Brayton power conversion system for a low volume, light weight marine propulsion plant."

The emphasis on low volume plant recognizes the sensitivity of containment and shielding weight, in potential nuclear plant applications, to powerplant volume. The integrated assembly minimizes the powerplant volume by eliminating the separation space between the various components of the system and the volume of interconnecting ducting and provisions for thermal expansion.

Other considerations which are pertinent to the feasibility of the integrated assembly concept are availability, reliability and maintainability. The need for compactness and the adoption of the integrated system, which effectively eliminates the free space around the various components of the system, requires that maintenance requirements be given special consideration. Means must be provided to obtain access to the components for maintenance or replacement when necessary. However, it must be recognized that the maintenance requirements cannot be considered independently of the system reliability and the required availability. Obviously an acceptable tradeoff between reliability and maintainability must be made to achieve the required plant availability. The turbomachinery is design to be capable of operating 40,000 hours without maintenance, consistent with the requirement for 10,000 EFPH at 25 percent duty cycle. The reliability required of the components is, nevertheless, no greater than that achieved in commercial combustion gas turbines which provide operating lives on this order without maintenance. Moreover this performance is obtained commercial combustion gas turbine in spite of relatively unreliable chemical combustor components which are not present in the CCCBS. In addition to the high reliability expected, the turbomachinery is designed to be easily removed, as a unit, from the integrated assembly for maintenance or replacement. Feasilibity of the integrated assembly from the standpoint of high plant availability is therefore doubly assured.

Removal of the turbomachinery from the powerplant is a straightforward operation. During this operation the rear section of

the pressure vessel is removed with the turbomachinery and functions as a lifting fixture, in the case of a horizontally mounted assembly. The center of gravity of the turbomachinery, generator and rear pressure vessel section is located approximately at the attachment flange of the rear section of the vessel to the center frame structure. The turbomachinery, generator and rear assembly can thus be supported at the center of gravity, from a crane or transporter device, while the unit is removed axially from the powerplant. Jacking screws in the attachment flange are used to start the flange separation. In a vertical assembly, the turbomachinery assembly would simply be lifted vertically from the powerplant while suspended from the centerline at the generator end. In a horizontally mounted nuclear installation, it is envisioned that the transporter would be mounted inside the containment and would transfer the turbomachinery to a suitably located penetration in the containment.

CCCBS feasibility is enhanced by the integrated assembly with respect to several other considerations. These include, the retention of helium within the system, low vulnerability to external influences and noise and vibration. The enveloping pressure vessel simplifies helium retention by greatly reducing the number and complexity of the component joints to be sealed. Some leakage is tolerable at inter-component joints within the system as this leakage remains confined within the system. Relatively simple means of accommodating flow path expansion, such as metal-to-metal sliding joints and O-rings, can be employed. The vessel also protects the internal components from damage and attenuates sound and vibration originating in the system.

Feasibility of the integrated system from the standpont of manufacturability and cost must also be considered. While the additional cost of the enveloping pressure vessel is incurred, the costs of complex heat exchanger headering and interconnecting ducting are minimized. Moreover, the elimination of interconnecting ducting removes the problem of accommodating ducting thermal expansion, involving the additional expense of metal bellows of questionable reliability. Vessel manufacture is straightforward and can employ low alloy steels to ASME code criteria, as a result of the low temperature environment mentioned earlier.

In assessing the variations in CCCBS capability which might be expected with changes in the integrated assembly requirement, (i.e., the adoption of separate components connected by ducting) the most important effect is an increase in plant internal volume. The volume increase would result simply from the additional volume of the ducting required to connect the components of the system. The resulting increase in the total gas volume contained by the plant also increases the required helium storage volume necessary to provide the inventory control capability between rated and 25 percent power. In addition to this increase in internal or contained gas volume of the plant, there is an additional increment of wasted volume between the components which

adversely affects the feasibility of the plant for incontainment application in a nuclear installation. These volume increases also result, in turn, in increased powerplant weight.

Another aspect of CCCBS performance which would be adversely affected by the adoption of separate components and interconnecting ducting is the cycle overall pressure drop. This is because the length and complexity of the inter-component flow paths are almost certain to be increased over those of the integrated assembly. The resulting increased pressure drop would reduce both the thermal efficiency and the specific power output of the plant, resulting in increased specific weight.

The capabilities of the CCCBS with respect to helium retention, vulnerability, noise and vibration would also be degraded if a separate component system with interconnecting ducting were to be adopted instead of the integrated assembly. Helium leakage would be more probable as a result of the increased number of component and ducting joints available as potential leakage sources. The interconnecting ducting and its expansion joint systems would be more vulnerable to damage from external sources, such as ship collisions and missiles, than would the enveloping pressure vessel. Noise and vibration originating in the turbomachinery would be more apparent in a separate component system, than when attenuated by the pressure vessel in an integrated assembly.

3. State Points and Component Efficiencies

The component efficiencies and state points specified in Table 3-2 (Section 3.4.1) were judged to be reasonable based on previous studies and demonstrated component state-of-the-art. Compressor and turbine efficiencies of 85 percent and 90 percent respectively were conservatively set at about 2 percent less than obtained in the MGCR program (Reference 3). Subsequent critical component evaluations performed by Westinghouse Combustion Turbines Division for CCCBS turbine and compressor components have indicated that these levels of efficiency should be easily achievable. Feasibility of the 927°C (1700°F) turbine inlet temperature and 10.3 MPa (1500 psia) pressure was shown in the discussion at the beginning of this section. The recuperator and cycle cooler effectiveness of 80 percent and 98 percent respectively have been shown to be feasible in the component evaluations. This subject is discussed in Section 8.2.1.

4. Heat Exchanger Envelope and Bearing Loads

The heat exchanger envelope and bearing load constraints were specified as second level requirements as a result of the scoping design work which resulted in the early trail design. These requirements were primarily intended for the guidance of the subcontractor in his initial engineering of the heat exchangers and bearings and to ensure the compatibility of the resulting components with the overall system.

The feasibility of the CCCBS with respect to these requirements has been demonstrated by successful engineering of heat exchangers and bearings which are compatible with the system and which can be accommodated within the space constraints. The bearing and heat exchanger evaluations are reported in Sections 8.1 and 8.2 respectively.

In considering the variations in expected capabilities of the CCCBS with changes in these requirements it must be emphasized that the envelope constraints were formulated consistent with the overall program objective of "a low volume, light weight marine propulsion plant." The specific envelope requirements were judged to be consistent with achieving a CCCBS specific weight of 1.22 Kg/KW (2 lb/SHP) a judgment which has been vindicated by the results of the study (see Section 3.4.1). However, changes in the envelope requirements will produce corresponding changes in the CCCBS capability.

If all envelope constraints are eliminated and, as a result, the cooler and recuperator components are allowed to increase in size, the specific weights for CCCBS and total powerplant with fuel will increase. As a result, the point will be reached where the 1.22 Kg/KW (2 lb/SHP) and 9.15 Kg/KW (15 lb/SHP) values for CCCBS and total powerplant with fuel cannot be met and the resulting powerplant no longer meets the overall program objective of "low volume and light weight" consistent with the SES application. Nevertheless the resulting powerplant may be quite suitable for other applications which can accept a heavier and less compact installation. These applications, especially if non-nuclear, might also justify a non-integrated assembly of the components.

Increases in permissible size of cooler and recuperator components can result in reduced pressure loss and increased effectiveness with corresponding improvements in overall thermal efficiency. As mentioned earlier, an increase in recuperator effectiveness from 80 percent to 85 percent results in, approximately, a 1.5 percent improvement in thermal efficiency in spite of a higher pressure drop assumption for the increased effectiveness recuperator. It is noteworthy that, as recuperator effectiveness is increased, the thermal efficiency optimizes at a lower cycle pressure ratio, suggesting that the increased cost of a larger recuperator might be compensated for by the reduced cost of lower pressure ratio turbomachinery.

If, on the other hand, the heat exchanger envelope requirements are changed to further restrict their volume, the demands for compactness will tend to force the heat exchangers towards lower effectiveness and higher pressure drop designs, resulting in reduced thermal efficiency. It may be possible to maintain effectiveness, in spite of volume reductions, by resort to finer heat exchanger geometry (greater surface area to volume ratio) but this will tend to increase costs. Certain high performance applications, demanding minimum volume and weight such as aircraft propulsion powerplants, could possibly justify substantially increased costs in return for reduced volume and weight.

3.5.3 FEASIBILITY WITH RESPECT TO OTHER CONSIDERATIONS

1. Helium Effects

The evaluation of CCCBS feasibility must consider the high heat transfer capability which, although generally advantageous in heat exchanger components, can produce high material temperature gradients and thermal stresses. This problem can arise as a steady state condition wherever hot and cold fluid streams are separated by a single thickness of material. The high heat transfer coefficients at the walls cause a relatively large portion of the overall temperature gradient to appear in the metal wall. Problems can also arise under transient conditions, where a single fluid stream impinges upon a substantial thickness of material. In this case the high heat transfer coefficient at the wall causes the temperature of the surface lamina of the material to closely follow the changing temperature of the fluid. The temperature of the material remote from the surface changes more slowly due to its thermal inertia which is a function of the material specific heat, density and conductivity. As a result, a relatively large portion of the overall fluid to metal temperature gradient appears in the metal.

Provision has been made in the CCCBS design to limit the metal temperature gradients and thermal stresses to acceptable values. Generally this has consisted of providing a relatively stagnant layer of helium to limit heat transfer flux. An example of this treatment, applied to the steady state problem arising from adjacent hot and cold streams, can be seen in the design of the turbine inlet plenum and hot gas duct from the heat source. A double walled construction is employed, the two walls being separated by a thin layer of stagnant gas. In this arrangement the outer wall performs the structural role of carrying the pressure difference loads and operates essentially at the temperature of the cold gas stream. The inner wall functions as a liner to protect the outer wall from the hot gas and operated, essentially unstressed, at the temperature of the hot gas. The major portion of the temperature gradient between the hot and cold streams is thus imposed on the stagnant gas gap, limiting temperature gradients and thermal stresses in the metal walls.

The potential transient thermal stress problem in thick walled components has been avoided in the CCCBS design by confining the hot gas streams, which are capable of experiencing large temperature transients, within essentially unstressed thin-walled ducts. An example of this approach can be seen in the gas path from the power turbine outlet to the recuperator modules. The turbine outlet gas, normally at a temperature of approximately 500°C (932°F) is confined within thin walled liners, ducts, and plena. The thick walled pressure vessel which envelops the powerplant is exposed at this location to the gas flow from the power turbine balance piston at approximately 150°C (302°F). Since the total range of temperature through which the pressure vessel operates, from full power to shutdown

is therefore limited to less than 150°C (302°F), the probability of inducing a substantial temperature gradient in the material is greatly reduced. A similar approach is used to confine the hot gas leaving the high pressure turbine and the gas flowing to the heat source.

Feasibility of the CCCBS could also be adversely affected by the propensity of helium to leak in significant quantities through relatively small gaps. Such gaps can exist at poorly fitted joints in turbomachinery plant, especially those which occur in split compressor and turbine casings at the intersection of the axial split plane with the end faces. These locations are notoriously difficult to seal positively and are invariably a source of chronic leakage. While these intersection joints could be eliminated by resorting to unsplit casings, this would adversely affect manufacturing feasibility by requiring the use of axial assembly methods and would greatly reduce access to the turbomachinery internals for inspection and maintenance.

The integrated assembly concept allows the use of conventional commercial gas turbine practice, with split casings, by providing a means for retaining the helium leakage within the system. The external pressure vessel forms the ultimate pressure boundary and is much more amenable to the application of positive sealing methods than is the turbomachinery. The joints in the pressure vessel have simple circular geometry and are at a relatively low temperature, allowing the use of elastomer O-ring static face type seals. Access to the turbomachinery for maintenance is provided by designing the turbomachinery to be axially removable from the powerplant assembly as a unit. Dynamic, radially sealing, O-rings are used at the turbomachinery-to-power-plant assembly flow path separation joints and these seals are capable of accommodating the relative axial motion at assembly and during the expansion and contraction of the turbomachinery. Some leakage can be tolerated at these seals because the leakage is confined to the system within the pressure vessel and merely represents a small loss of efficiency.

Thus the helium characteristics tending to cause thermal stresses and excessive leakage can be easily accommodated by the integrated assembly concept without precluding access to the turbomachinery for inspection and maintenance.

2. Heat Source

The CCCBS can use either a nuclear or fossile heat source without any major changes needed in its design. The choice of the heat source would affect the CCCBS design only in a secondary manner, in items such as nuclear shielding and containment.

The ability of the plant to respond to any control inputs should not be restricted by the response of the CCCBS. With a fossil heat source, the limiting factor would be the allowable thermal stress in the heat exchanger tubing. In a nuclear heat source, as shown in

Section 6.1.2, the required plug shield at the reactor outlet would tend to reduce any temperature ramp that would be felt at the inlet of the cycle turbine.

3. Survivability in the Event of Malfunctions

CCCBS feasibility is enhanced if the powerplant is able to survive conceivable malfunctions without damage. One such malfunction analyzed in detail, is the complete loss of load such as might occur, in an electrical transmission system, through the tripping of circuit breakers or in a mechanical drive system, through a shafting, coupling or gear failure.

The primary concern in the event of a complete loss of load is the capability of the free power turbine to withstand the centrifugal stresses at the overspeed condition reached after the loss of load. Analysis has shown that an overspeed of about 80 percent (over the normal power turbine operating speed of 9000 RPM) may be reached by using helium inventory conctorl and an overspeed on which energy source is used. This overspeed is insensitive to the actuating time and size of the inventory control valve, since it is primarily generated by the plant conditions that exist below 25 percent output power where helium inventory control is not used. By shifting the turbine maximum efficiency point to a speed lower than the rated speed, the overspeed can be limited to about 60 percent. In addition, a turbine bypass control could be used to bypass helium flow around the power turbine and, if necessary, reduce the peak overspeed still further.

A stress analysis of the most highly stressed (last stage) power turbine disc, neglecting thermal stresses, revealed an average tangential stress of 276 MPa (40,039 psi) with a peak tangential stress at the bore hole of 371 MPa (53,827 psi) at the 9000 RPM normal operating speed. At 60 percent overspeed the bore stress value increases to 950 MPa (137,797 psi). This value approximates the yield strength capability of candidate disc materials such as AISI 4340 and Inconel 718 and implies little or no yielding at the most highly stressed bore location. Thus, the power turbine overspeed capability under complete loss of load conditions is adequate for the limited number of loss of load cycles anticipated in the powerplant life.

This evaluation is conservative in a number of respects. The onset of yielding at the disc bore does not imply imminent failure since yielding is limited by the adjacent lower stressed material and merely results in a redistribution of stress further out in the disc. Also, the disc design analyzed is not yet optimized and could, if required, be substantially strengthened by thickening and by the use of a smaller bore hole. The need for such a redesign to improve the survivability of an overspeed condition has not yet been demonstrated.

Other effects of the complete loss of load indicated by the analysis were a change in the pressure difference from the high to low pressure side of the recuperator from 8.3 MPa (1210 psia) to 4.4 MPa (640 psia) in approximately 0.8 seconds. This was accompanied by an increase in the temperature on the low pressure side of the recuperator from 527°C (980°F) to almost 704°C (1300°F) resulting in an increase in the temperature difference between the high and low pressure sides of the recuperator from 67°C (120°F) to approximately 244°C (440°F) in 0.8 seconds. However, the vast majority of this temperature difference is felt in the helium film temperature drop in going from the bulk of the fluid to the wall. The temperature difference through the tube wall would increase from 3°C (5°F) to 6°C (10°F) in this time, a negligible amount from a thermal stress standpoint.

The thrust bearing loads on the turbocompressor and power turbine bearings were also found to be affected by complete loss of load. The power turbine thrust load reduced from 57,000 N (12,800 lb_f) to 22,240 N (5,000 lb_f) in 0.6 seconds, which was obviously acceptable. The turbocompressor thrust (-200 lb_f) to over 7117 N (1600 lb_f) in the same period. The latter value is however relatively small in relation to the bearing capability of ± 44,480 N (10,000 lb_f).

The capability of the CCCBS to withstand several cycles of complete loss of load without significant damage is believed to be representative of the ruggedness of the powerplant and provides some evidence of its feasibility with respect to survivability in the event of a malfunction.

4. Two-Unit Operation in Parallel

In a nuclear installation, the use of two in-parallel units which can be operated independently or together provides increased assurance against a loss of coolant accident due to failure of one of the turbomachinery units.

The operation of a single unit of a two-unit parallel system with the other unit shut down imposes conditions on the shut down unit which are not experienced during normal operation. Because the units supply and receive gas at the high pressure point of the cycle, the shutdown unit will experience high pressure throughout. Since the pressure in the reactor supply annulus of the operating unit is higher than in the turbine supply pipe, gas will tend to flow in the reverse direction through the shut down unit. This is prevented by check valves in the reactor supply flow path which close and prevent the reverse flow. Pressure in the shut down unit is then maintained at the turbine inlet pressure of the operating unit.

Feasibility of the CCCBS for two-unit operation requires that the pressure vessel be capable of containing the full system pressure and this capability has been provided in the third definition design concept illustrated in Figure 7-1. As discussed in Section 3.5.2.1 the powerplant specific weight of 0.82 kG/kW (1.34 lb/SHP) is increased

as a result of providing two-unit operation capability, from the lower value of 0.65 kG/kW (1.07 lb/SHP), which would be achievable in a powerplant designed for single-unit operation only.

Another aspect of CCCBS feasibility with respect to two-unit operation, is the effect on the shut down unit of the reverse flow of check valve leakage gas through the system. The leakage gas originates from the recuperator high pressure outlet of the operating unit and is at a temperature of 453°C (847°F) at rated power. This gas is cooled in passing through the recuperator modules by the cool gas passing in the other direction on the other side of the tubes. The cool gas has previously passed through both the precooler and intercooler, which are assumed to be receiving a normal water coolant flow. Therefore, the check valve leakage gas entering the recuperator modules at 453°C (847°F) will be cooled to a temperature well below the elastomer O-ring capability before it reaches the O-rings at the cold ends of the recuperator. This gas then passes into the center frame and enters the turbomachinery at the high pressure compressor outlet. The gas then passes in the reverse direction through the compressor, is further cooled in the intercooler and subsequently passes in the reverse direction through the low pressure compressor. Finally, the cooled leakage gas passes through the precooler, the recuperator, the power turbine and the high pressure turbine in the reverse direction and returns to the operating unit through the turbine supply pipe. Therefore, due to the cooling which the leakage gas receives in the recuperator modules immediately after leaving the check valves, no damage will be caused to any of the elastomer O-rings in the system.

5. Part Load Operation

CCCBS power control is achieved by a system of helium inventory control between rated and 25 percent power. Below 25 percent power, control is achieved by varying turbine inlet temperature at constant inventory. Control of the system is accomplished by automatic synchronized control if turbine inlet temperature and of system pressure. Demanded turbine inlet temperature is varied linearly from 927°C (1700°F) at full power to 899°C (1650°F) at 25 percent power to maintain 10,000 EFPH life capability.

System pressure, and therefore power, is controlled by adjusting the quantity of helium within the primary flow system. Helium is bled from the high pressure compressor exit into the control gas storage bottles to reduce power output. In order to increase power output, helium is beld from the storage bottles to the low pressure compressor inlet. The control gas storage bottles are sized and sequenced such that no pumping of the stored helium is required.

The control concept described above has the advantages of maintaining all system temperatures essentially constant over the normal operating range, regardless of whether the output shaft speed is constant or variable. Since the temperatures are essentially constant, transient

thermal stresses are minimized. In addition, the gas generator turbomachinery (compressors and high pressure turbine) operate at essentially constant speed over the normal power range. Part-power efficiencies are therefore maintained at close to rated condition values, and the turbomachinery is required to operate over only a small range of conditions.

Feasibility of this system has been demonstrated by the results of the transient analysis performed during the study and discussed in Section 6.1.2. From these results it would appear that the reference control scheme (helium inventory and reactor outlet temperature control) can satisfactorily control normal operating transients over the entire operating range. Also, it has the advantage of resulting in essentially constant station temperatures and plant efficiency over the normal operating range (25 percent to 100 percent output power).

6. Vulnerability to External Influences

The CCCBS in Navy propulsion powerplant applications should provide a high degree of resistance to damage resulting from collision, missiles and other external influences. In this respect the integrated assembly is particularly advantageous in that the components of the system are protected by the external pressure vessel. The elimination of exposed interconnecting ducting and expansion joints, which substantially increase the susceptibility of separate-component systems to foreign object damage, greatly reduces the vulnerability of the integrated assembly.

In a nuclear installation, the compactness achieved by the integrated assembly allows the reactor and two of the CCCBS units to be accommodated within a shielded containment vessel to achieve a total system specific weight of less than 9.1 kG/kW (15 lb/SHP) while providing the capability of surviving a 30 knot ship collision and 1000 foot deep submergence without loss of containment integrity.

7. Installation Compatibility

The enveloping pressure vessel of the integrated assembly provides the equivalent of a baseplate for the powerplant components. As a result the rotating system components are well supported in a completely aligned relationship with one another. Moreover, support and alignment are maintained regardless of any flexing of the ship structure. In those installations utilizing superconducting generators and electrical power transmission systems, the necessity for accurate alignment of the powerplant with the ship's power transmission equipment is completely eliminated. As a result, the CCCBS in this form is particularly suitable for use in flexible hull structures, such as those of lightweight high speed ships.

Services to and from the CCCBS, such as cooling water, electrical power transmission and gas bearing supplies can be easily designed with the necessary flexibility to accommodate relative motions between the powerplant and ship. This is particularly true in a nuclear installation

where the reactor and CCCBS units are accommodated within a containment vessel penetrated only by relatively cool water, electrical and gas lines. In a fossil fuel installation the situation is somewhat complicated by the heat source-to-CCCBS working fluid connections which convey hot helium and which must be designed to accommodate thermal expansion and hull motions. However, several design approaches are possible, depending on the compactness of the heat source and the requirements of the installation.

The external pressure vessel of the integrated assembly is exposed only to relatively low temperature helium at a maximum temperature of approximately 150°C (302°F). As a result, thermal expansion motions are minimized and the mounting of the powerplant assembly in the ship and the requirements for thermal protection and insulation are simplified.

The CCCBS is also compatible with mechanical drive installations. In evaluating the feasibility of a mechanical drive, a power turbine with a design output speed of 36000 RPM was defined. The required sealing provisions for the drive shaft penetration of the pressure boundary were developed. These provisions were also evaluated for the 9000 RPM turbine and found to be feasible. Reduction gearing having 9000 RPM input and 3600 RPM output speeds was also examined and found to provide a feasible alternative to the direct drive turbine designs. The CCCBS is therefore judged to be feasible for both electrical and mechanical drive installations.

8. Noise and Vibration

A desirable feature of a Navy combat vessel is as low a noise and vibration level as possible. The integrated assembly CCCBS concept with close-coupled superconducting generator and electrical power transmission system should result in very low noise and vibration levels as a result of the elimination of gearing and the enclosure of the turbomachinery within the developing pressure vessel. Also, the elimination of the need for precise alignment or location of the assembly allows the use of low spring rate mounts capable of isolating high frequency vibrations from the hull. The low operating temperature of the vessel also simplifies the application of accoustic insulation material to the vessel if required.

9. Availability/Reliability/Maintainability

System availability refers to a combination of reliability and maintainability. A simplified formula for overall availability shows that with no down time the system availability is 1.0 (or 100% of the time). Realistically the availability is maximized by making accessible components highly maintainable and by making those components whose accessibility is limited highly reliable and redundant.

In the case of a nuclear application of the CCCBS the constraints on maintenance (i.e., some maintenance actions are of long duration because of the containment of certain components) dictate that the effects of long down times be reduced by making them very rare occurrences. This is accomplished by system design to minimize the number of contained critical items and designing for high reliability those other components which must be contained.

Maintenance capability falls into the classes shown in Table 3-3. It is assumed that a specific port facility is located on each coast. This assumption can be trade studied against the practicality of the Panama Canal passage and the time and cost of the additional travel to a single facility. A maintenance tender is also a possibility. Further, replacement should be practical in any port where an adequate crane is available. It is further assumed that each facility would maintain at least one spare unit in ready status. The practicality of complete replacement, of the containment and included system, in a ten day port period is readily envisaged and is assumed in this study as a requirement.

Pending vehicle design, it is impractical to exactly define the degree of on-board maintenance capability. At this time, it is assumed that the capability of weight sensitive, high speed vehicles such as the SES will preclude the incorporation of extensive machine shops, maintenance equipment and spare parts of the sort presently included in naval displacement ships. Further, it is recognized that on-board maintenance capability may be a function of vehicle size, i.e., increasingly practical as the vehicle size increases. For the purpose of this design study the assumption has been made that major maintenance (generators, turbines, compressors and heat exchangers) can be accomplished only in the port facility.

Auxiliary systems for which some maintenance is necessary are located external to the containment vessel for ease of access. This includes the gas bearing auxiliary helium supply systems, electronic controls, cyrogenic helium systems, power system switchgear, and all water system pumps and water treatment components. Of course, the propulsion and lift motors and seawater heat exchangers must be located separate from the powerplant.

It should be noted that the CCCBS concept lends itself to design for high operational reliability for the reasons indicated in Table 3-4. The judicious use of modern reliability and product assurance techniques in the development, design, and construction of these systems can be expected to provide overall powerplant availability adequate to meet the user requirements. These techniques represent one more example of the benefits to be realized from the United States space program where very high component reliability has been consistently demonstrated.

TABLE 3-3

DESIGN GUIDE FOR MAINTAINABILITY

OPERATIONAL STATUS	INSIDE PRESSURE VESSEL	OUTSIDE PRESSURE VESSEL
Underway and Non-Special Port	Preventive - None	Preventive - None
	Corrective - None	Corrective - Selective
		(Extensive in port if spare parts available)
Special Port or Tender	Preventive - None	Preventive - Extensive
	Corrective - Selective	Corrective - Extensive
Base Port*	Preventive - None	Normal Overhaul Activities
	*Corrective - Selective	

*Replacement of complete system is possible.

TABLE 3-4

SOME CHARACTERISTICS FAVORING RELIABLE OPERATIONS

- Chemically inert working fluid.
- Minimum and predictable containments.
- Moderate thermal and mechanical stresses.
- Managed or programmed load change rates.
- Normal access to most auxiliary systems.
- Limited number of inaccessible components.
- Redundancy of critical components.

In evaluating the availability capabilities of the CCCBS, the turbomachinery package has been identified to be the most critical assembly. However, an availability capability of 0.99 will impose a MTBF requirement on the turbomachinery assembly not much higher than requirements for a similar lightweight air cycle gas turbine in the DD-963 destroyer (15,000 hours vs. 10,000 hours for the DD-963). It must be recognized that most air cycle turbomachinery hot gas path exposure is less severe than comparable air gas turbines because:

a) Turbine inlet radial temperature profile is significantly flattened in CCCBS plants with either reactor or fossile heat sources by the elimination of the need for combustor bypass cooling.

b) Turbine inlet hot spots due to combustor and injector malfunctions are reduced due to additional heat capacity of the reactor plug shield.

c) Controlled turbine inlet temperature minimizes thermal transients.

Thus, based on the considerations discussed above, the CCCBS is judged to be feasible and practical in meeting the most stringent reliability requirements of the nuclear application.

In a fossil fuel application, containment will not be required and, as a result, the opportunities for direct maintenance on the turbomachinery components will be greater, allowing some relaxation in the reliability required to achieve a specified availability. Consistent with these less stringent conditions the CCCBS integrated assembly has been designed to facilitate the removal of the turbomachinery as a unit, from the powerplant. This is achieved by removing the bolts attaching the rear pressure vessel section to the center frame structure. The turbomachinery and generator can then be withdrawn axially, as a unit, suspended from its center of gravity near the rear pressure vessel attachment flange. Obviously, sufficient clear space must be available adjacent to the powerplant assembly, into which the turbomachinery assembly can be withdrawn. Furthermore, if the turbomachinery is to be removed from the ship for inspection and/or maintenance, suitable hatchway access must be provided. Alternatively if on-board maintenance capability is to be provided, its withdrawal space adjacent to the powerplant could be utilized for this purpose. After transfer of the turbomachinery to the maintenance area, access to the hot components, blading and bearings can be obtained by removing the appropriate split casing components in accordance with standard commercial gas turbine practice.

While the turbomachinery removal facility discussed above is envisaged primarily as a fossil fuel powerplant feature, it appears feasible that the high fission product retention expected in state-of-the-art graphite core reactors would allow this facility to be exploited in a nuclear plant. In such a case, the turbomachinery could be supported during removal from the powerplant, on a transporter device inside the

containment which would transfer the turbomachinery through a penetration in the containment to an external crane and lifting fixture. Obviously such a procedure would be employed only under the most strict controls and with monitoring for radioactive contamination.

Feasibility of the CCCBS with respect to availability/reliability/maintainability is believed to have been adequately demonstrated by the work performed to date for this study. Sufficient design flexibility is available to achieve an appropriate combination of reliability and maintainability in either nuclear or fossil plants to meet the expected availability requirements.

10. Manufacturability and Cost

Manufacture of the CCCBS components is straightforward and requires only conventional well established manufacturing practices. The turbomachinery is conservatively designed and, although sufficiently lightweight, employs disc and casing components which are not so thin as to require special care and support during machining. The rotor construction proposed in the third definition design concept employs stacked discs with curvic-couplings and through-bolts, an arrangement which has been widely used in gas turbines manufactured by Westinghouse and others. The split casing construction follows conventional commercial gas turbine practice and can employ either cast or fabricated components. Uncooled turbine blades made of IN 100, a widely used superalloy material, are used in the first stage turbine, simplifying manufacture. Refractory materials or complex cooled blade designs are not required to achieve the conservatively chosen 927°C (1700°F) turbine inlet temperature.

As a result of the relatively simple and conventional design, the turbomachinery component costs are expected to be comparable with those of similar commercial gas turbines. However, the relatively large number of stages characteristic of helium turbomachinery results in increased numbers of parts and correspondingly increased costs. Nevertheless, CCCBS turbomachinery costs, on a per-horsepower basis, are expected to be less than in current open-cycle gas turbines as a result of the greatly reduced size of the helium turbomachinery possible with the high system pressures.

The coolers are primarily an improvement over another helical design concept built for the Brayton Heat Exchanger Unit Alternate Design (BHXU-Alternate) documented in Reference 6. These units were successfully built and tested, and appeared able to survive in the shock environment that the CCCBS was designed to meet. By building on the knowledge gained from the BHXU project, there does not appear to be any reason why a helical design cooler could not be successfully built for CCCBS.

Heat exchangers proposed for the CCCBS represent state-of-the-art technology. The helical finned-tube precooler and intercooler achieve high effectiveness with a minimum number of tubes (~4500 in each). As a result, the number of tube joints is also minimized, benefitting reliability and cost. In the coolers, reliability is extremely important, since the failure of a tube joint results in leakage of the helium cooling fluid into the cooling water system. The recuperator, on the otherhand, is less critical from the standpoint of leakage, since helium is merely transferred through the leak to the low pressure side of the recuperator and remains confined to the system. A furnace brazed assembly of the approximately 10,000 tubes in each recuperator module is believed to be an entirely feasible manufacturing approach which can be expected to achieve adequate reliability at an acceptable cost. By forming the tube ends into a hexagonal shape, the tubes can be directly brazed together at their ends, eliminating the need for tube sheets with large numbers of accurately drilled holes. Brazing defects can be rebrazed during subsequent furnace brazing cycles.

Although there does not appear to be any reason at present why a successful gas bearing design is not possible, as an alternative oil lubricated bearing technology can be substituted to provide a viable, though less compatible colution. This would allow a prototype engine to be developed without having the bearing development severely impact the engine program schedule. Once a successful gas bearing was developed, it could be incoporated into the design during the qualification phase of the program.

The helium environment with its reduced oxidation potential and the moderate tube temperature of approximately 500°C (932°F) may allow the use of low alloy steel tube material such as 2-1/4 Cr-1 Mo instead of the more expensive stainless and superalloy material, further reducing recuperator costs.

The external pressure vessel and center frame components are simple welded structures fabricated from SA 533 low alloy steel in accordance with the ASME pressure vessel code. This material is used in the pressure vessels of Westinghouse pressurized water reactors. The maximum metal thickness of approximately 51 mm (3 in.) is well within the fabrication state-of-the-art. The maximum vessel operating temperature is approximately 200°C (392°F) allowing the use of elastomer O-ring seals at bolted flanges and penetrations. Hot gas flowing from the power turbine outlet to the recuperator and from the heat source to the HP turbine inlet is confined within essentially unstressed thin walled liners, thus minimizing the use of costly high temperature material.

Thus, the CCCBS manufacturability is judged to be well within the present state-of-the-art and should not prove to be unduly costly on a dollars per horsepower basis.

11. Materials and Technology Availability

In general, the CCCBS is well within the present materials and technology state-of-the-art. No technological breakthroughs are needed to ensure feasibility. A hardware design and development program started forthwith would not be expected to encounter serious problems. There are, however, certain areas of the CCCBS as defined in the third definition powerplant concept, which require mechanical development and test work if the highest potential performance of the powerplant is to be achieved.

Gas bearings were selected for their superior compatibility with the closed cycle plant but with the recognition that experience to date had been limited to much smaller turbomachinery units. As discussed in Section 7.0, further design and development is necessary to provide gas bearings with the required shock capability.

Another area of CCCBS technology which could be improved by the expenditure of development effort is the design data base in the mechanical and structural properties of materials in helium. The identification of suitable abradable materials for use in turbine and compressor shrouds and in labyrinth seal stationary members is a worthwhile development goal. Also important is the development of surface treatments to resist the tendency of mating surface to bond and gall in a clean helium environment.

Turbomachinery developments which take advantage of the high acoustic velocity in helium to increase the stage work capability while maintaining efficiency are also desirable. Some work of this type is already underway on high reaction compressor stages and, if successful, could increase the work per stage and allow the number of stages to be reduced.

Therefore, although materials and technology availability is sufficient at the present time to ensure feasibility, the developments discussed above are nevertheless desirable to achieve the full CCCBS potential.

12. Control and Protection

Of fundamental importance to concept feasibility is the ability to control and to protect the system under both normal and abnormal conditions. For this study, the CCCBS has been designed to be capable of being automatically controlled from a single station with a single demand parameter. Manual controls will also be provided for each control device. All of the control electronics will be located externally for accessibility.

For the purpose of feasibility evaluation, the requirements shown in Table 3-5 have been assumed for the SES application. These requirements are also similar to the requirements of other potential CCCBS applications. This study has defined reference control functions to fulfill these requirements. Confidence in the feasibility and

TABLE 3-5

CONTROL SYSTEM REQUIREMENTS

- Normal control range from 25 to 100 percent of full power. Control to be automatic with a signal from a single control station.
- Rate of change of power up to 10 percent of full power per second.
- Automatic controls for start-up and cooldown to be provided.
- Local control station to include both automatic and manual control capability.
- No single credible malfunction shall leave the power plant non-restartable.
- The control and protection system shall make maximum practical use of diverse redundancy of sensors to maximize system reliability.
- The control and protection system shall be designed for high reliability with minimum maintenance. However, control system electronics shall be accessible for maintenance.
- The order of preference for backup control modes shall be as follows:
    1. Automatic control
    2. Limiters which maintain the system in normal automatic control
    3. Overrides which transfer to backup control modes
    4. Manual
    5. Scram (in nuclear installations)
- All control functions shall be testable to assure proper functioning

adequacy of these control concepts is derived from the fact that only concepts already proven in other programs have been selected.

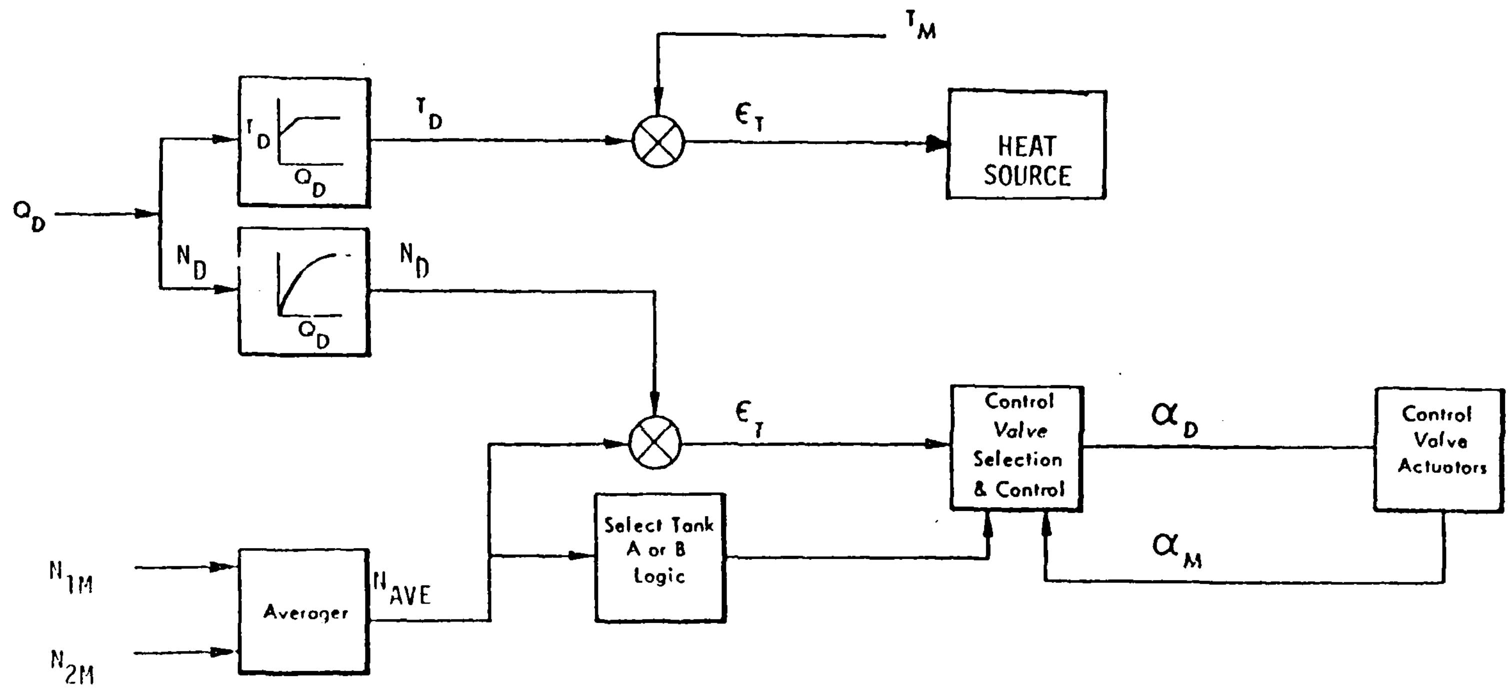
For this study, the CCCBS has been designed to be automatically controlled from a single station with a single demand. This demand could be a power demand direct from a throttle or could be a power demand generated from a summation of each of the individual load demands. All of the necessary powerplant control signals will then be generated from the one demand signal received by the control system.

The control concept includes control of system temperature at the heat source and control of system pressure by adding or removing helium working fluid into or from the primary system. The combination of the two control systems provides simple but highly responsive control of the CCCBS.

For the CCCBS representative design, heat source exit gas temperature is varied linearly from 927°C (1700°F) at rated power to 899°C (1650°F) at 25 percent power to maintain 10,000 EFPH capability. This almost constant temperature minimizes thermal transients and provides the capability for even faster than required transient capability since changes in component stored heat are minimized. In addition, each local pressure is essentially proportional to output power. Below 25 percent power, power is controlled by adjusting the heat source exit gas temperature only. The control concept for the CCCBS thus provides the versatility to allow optimization to meet specific requirements without changing the basic concept.

Figure 3-17 is a top level functional block diagram of the preliminary CCCBS control concept. While the specifics of the inputs which will be used to generate the CCCBS demands are dependent upon the operational requirements of the application, the system of Figure 3-17 illustrates the concept. A plant power demand is generated as a function of the vehicle measured and/or demanded power. The power demand to the CCCBS control system is used to provide synchronized demands for both of the two main CCCBS controls (temperature and pressure).

The demanded heat source outlet temperature or turbine temperature is compared to the measured temperature. An error between demanded and measured temperature will cause the heat source temperature control to move adjusting its outlet temperature. For this control parameter, as for other critical control parameters, multiple sensors will be used and their signals will be averaged and any failed sensors rejected. This average and reject function is necessary both to insure proper control and to allow for inclusion of redundancy in sensors for reliability. Experience has also shown that parameters such as heat source outlet temperature can be accurately determined not only by direct measurement but also by inference from other heat source temperatures, thus providing additional backups.



Parameters

Q - Power
N - Power Turbine Speed
ε - Error
α - Control Valve Position

Subscripts

D - Demand
M - Measured
T - Sum (demand-measured)

Figure 3-17. Preliminary CCCBS Control Concept

Since temperatures from the heat source exit to the turbine inlet are close to the same value under steady-state conditions, several locations are possible for sensing of measured temperature. However, there are differences in dynamic response, especially in a nuclear system, because of the energy stored in the plug shield. In this case, the reactor exit plenum is the preferred location for sensing of the control temperature, but other stations can also be measured both as backup control parameters and as monitors. In addition, in-core material temperatures can be measured using techniques developed in NERVA for diagnostic and backup control purposes. NERVA control systems tests conclusively demonstrated that either in-core material or reactor exit fluid temperature could be used for reactor control.

The demand power turbine speed is likewise compared against the measured speed. This speed error is then used to generate an inventory demand signal. If the demand speed is greater than the measured, helium is added to the primary system at the precooler inlet. If the demanded speed is less than the measured, helium is removed from the primary system by bleeding from the HPC exit. The use of bleed from the HPC exit and fill at the precooler inlet combined with proper sizing of the control valves permits power changes at the rate of ±10 percent per second over the range of 20 to 100 percent of power without the necessity for a separate control gas pumping system and without in-line throttle valves.

Not shown on Figure 3-17 are the pressure measurements taken in the low and high pressure control bottles, and at the low pressure recuperator exit and the high pressure compressor exit plena. These measurements are used to select whether bottle A or bottle B will be used for control purposes (bottle A is used below 45 percent power, while bottle B is used above 45 percent power, with a small overlap at the transition point).

Pressures in the system and in the control bottles as a function of power are shown in Figure 3-18. Two control bottles are required so that transients over the full range can be accomplished without pumping. (The source pressure, HPC exit, at 20 percent power is lower than the sink pressure, LPC inlet, at 100 percent). The bottles must be sized and the changeover pressure between the two bottles selected such that bottle volumes can accept the required inflow and provide the pressure necessary for the required outflow. A range of transition pressures for switching between bottles A and B is possible. For this system, a transition pressure of 45 percent permits the minimum total bottle volumes.

System control by changing the quantity of helium in the primary flow system has several advantages which has led to its selection as the control concept for the representative design. Control can be accomplished without in-line valves. This is desirable both from part-power efficiency considerations and from consideration of effects of value failures since in-line valves can fail such as

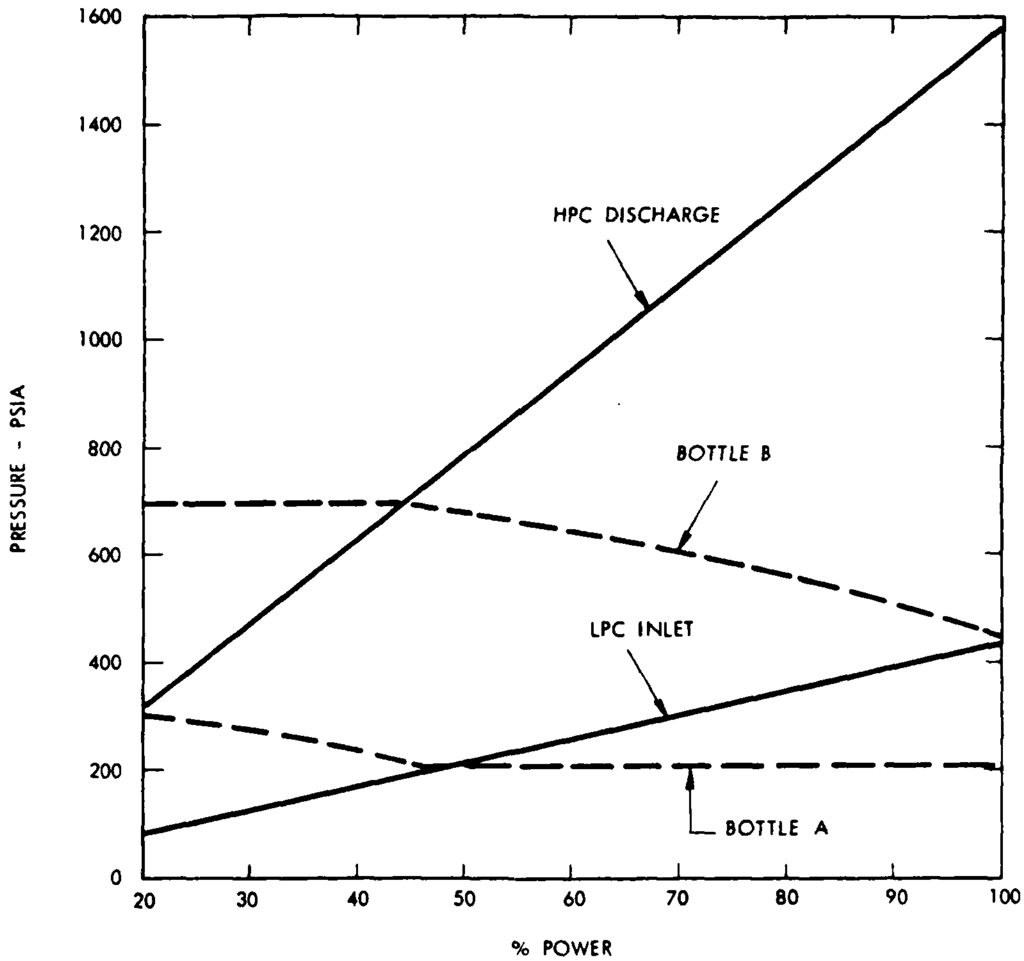


Figure 3-18. System and Control Bottle Pressures

to partially or completely block the flow. High levels of redundancy can be incorporated for bleed and fill valves and, even if complete failure of valve system occurs, an intermediate power level can be maintained. Of course, some weight penalty is incurred because of the storage bottles required. Alternate control methods are possible. The prime alternate is that of controlled bypass of the power turbine. In this alternate, flow would be extracted from the plenum between the HPT exit and the LPT inlet, throttled, and returned at the LPT exit thus partially bypassing the LPT.

Also shown in Figure 3-18 is an automatic control system to minimize speed and power mismatch between the two sets of turbomachinery. However, since mismatches should only occur because of hardware tolerances, the mismatches should be relatively constant over the normal operating range. In this case, the matching of the two systems will be accomplished through remote manual operations and the automatic control system could be deleted. Control and protection of the CCCBS under all conditions of operation is therefore judged to be feasible.

13. Development Required

As already discussed under the heating of "Materials and Technology Available"; the CCCBS is feasible using presently available materials and technology. However, as was pointed out, further gas bearing development is needed to extend the technology base to the larger rotor system of the CCCBS and provide the capability of meeting navy shock requirements if it is decided to incorporate these bearings in the CCCBS.

Materials property testing in helium is also desirable to extend the design data base. This should include friction and wear investigation as well as structural properties testing.

Turbomachinery development directed towards the better exploitation of favorable characteristics of helium, such as its high acoustic velocity, could possibly result in the achievement of higher stage loadings which, in turn, would allow the number of stages to be reduced.

However, these developments must be classed as desirable rather than mandatory and are not essential to CCCBS feasibility.

The research and development program recommended for the CCCBS is discussed in more detail in Section 4.

3.6 REFERENCES

1. "Convection in the Closed Brayton Cycle"; M.F. Taylor et. al.; Aero Mechanical Engineering Department, University of Arizona; Report No. 1248-3; April 1976.

2. "Heat and Momentum Transfer to Internal, Turbulent Flow of Helium-Argon Mixtures in Circular Tubes"; Aero Mechanical Engineering Department, University of Arizona, July 1976.

3. "Maritime Gas Cooled Reactor Project - Development of Turbomachinery," U.S. Maritime Administration Contract MA-2796, Engineering Report No. EC-167, Westinghouse Electric Corporation, June, 1963.

4. "Annual Review of the High Temperature Metals Research for VHTR at JAERI," Tatsuo Koudo, Chief, Materials Engineering Lab, Tokdi Research Establishment, JAERI, January, 1977.

5. "Shock Design Criteria for Surface Ships," NAVSEA 0908-LP-000-3010, May, 1976.

6. "Brayton Heat Exchanger Unit Development Program (Alternate Design), Final Design Report," J.D. Duncan, NASA CR-121269, August, 1973.

4.0 DEVELOPMENT PROGRAM

The construction of a production CCCBS plant requires a large amount of design effort to be spent in a number of different technology areas. This effort would include research and development on the various plant components, such as high temperature materials, compact heat exchangers, turbomachinery, and gas journal and thrust bearings. Each of these components have their own distinct technological problems that need to be solved. The establishment of a development program is required to enable these problems to be resolved in a reasonable fashion, and to allow for the integration of the entire plant to proceed in a timely manner.

A discussion of the development program needed for the design and integration of the entire CCCBS plant is given in Section 4.1. This program would be based upon the individual component and material development programs, discussed in Sections 4.2 and 4.3.

4.1 OVERALL CCCBS

The successful and timely completion of a production CCCBS powerplant requires that a development program be defined early in the project. This program would be a layout of the tasks to be completed during the design, development, construction, and test phases of the program.

An estimate of the schedule for the CCCBS program is shown in Figure 4-1. This schedule is assumed to start at the conclusion of the feasibility study documented in this report. A specific application for the CCCBS is assumed to have been defined, and the requirements on the plant have been specified.

The general schedule for the overall plant development would be as follows:

1) Preliminary system study and design for a prototype engine

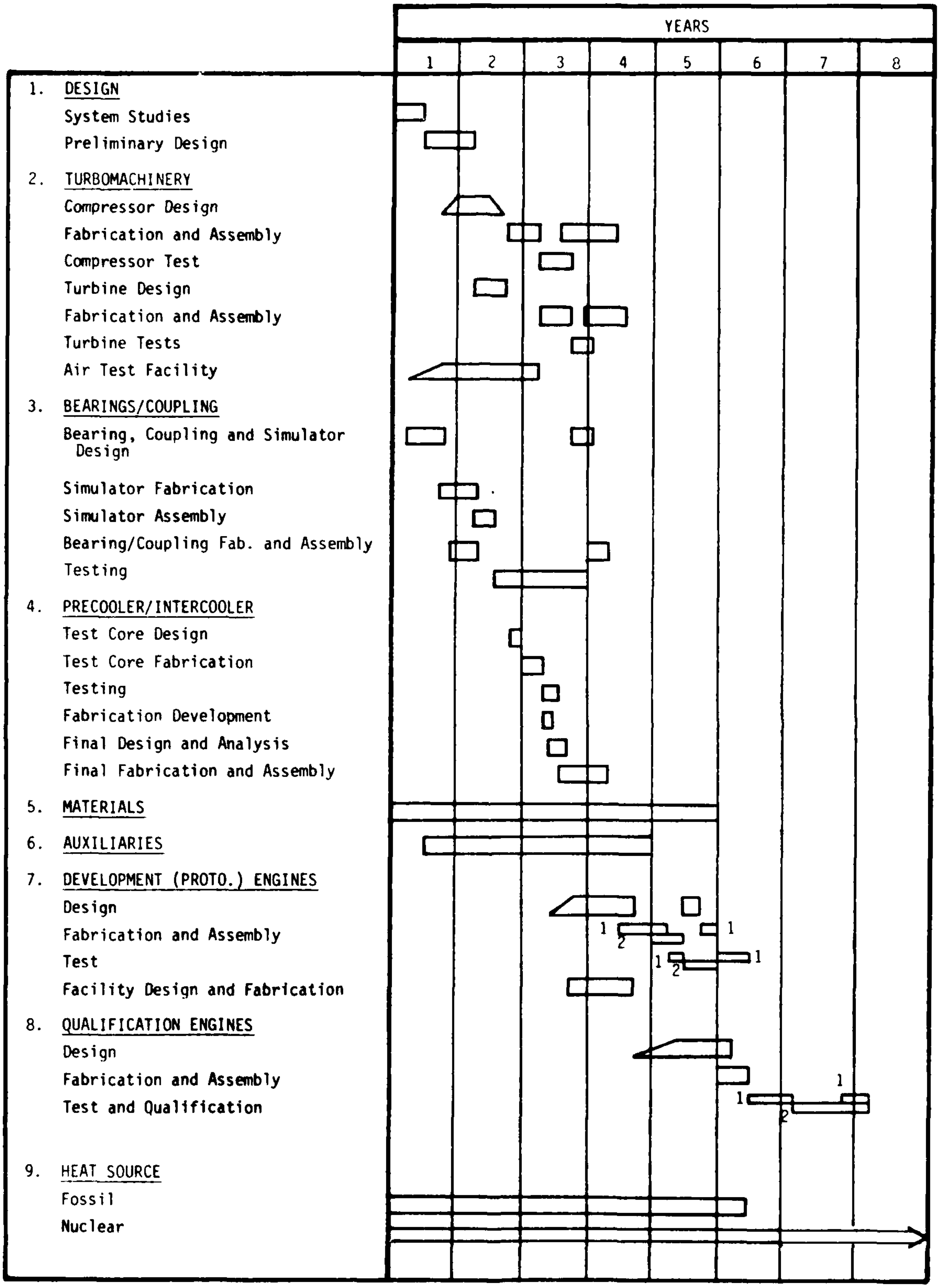


Figure 4-1. Overall CCCBS Development Schedule

2) Definition of development needed on individual components

3) Individual development and testing of representative components

4) Design, integration, assembly and testing of two prototype engines

5) Design, integration, assembly and testing of two qualification engines.

The two prototype engines would be full-scale versions of the production (or qualification) plants. These would be the first units to be tested in helium, and would be used to verify the design and analytical efforts. Following the successful testing of these prototype units, the two qualification engines would be built and tested. These engines would be used to conduct the military qualifications tests necessary to insure that the powerplant meets all of the performance and survivability requirements set forth at the start of the program.

It should be noted that there are no military specifications in existence today for closed-cycle Brayton powerplants that the manufacturer must meet. In order to avoid severely impacting the development schedule, these specifications should be available early in the design phase of the plant. Detailed component development would begin as early as 6 months into the project, and these specifications would be required by this time. Realizing the extent of the coverage of these specifications, their generation should commence as soon as possible after the need for a CCCBS powerplant is recognized.

The entire development program, from the definition of the powerplant application and performance requirements, through the qualification of the assembled plant, is assumed to take approximately 7 years. The individual tasks listed in Figure 4-1 are described below. For the individual components (Tasks 2, 3 and 4), detailed descriptions of their development programs are given in Section 4.2. The materials development program (Task 5) is defined in Section 4.3.

4.1.1 SYSTEM PRELIMINARY DESIGN

Following the declaration of the specific application for and requirements on the CCCBS, a number of system studies would have to be initiated. These would include cycle studies to determine the most likely type of Brayton cycle that

would best meet the needs of the specific plant application. Representative statepoints and plant component layouts would also be developed. The feasibility of the design would be demonstrated from the standpoint of materials, fabrication, and component and overall plant performance.

One of the prime purposes of the preliminary design would be to determine those technical areas that require additional development. These would primarily include the turbomachinery, gas bearings, turbocompressor shaft coupling, and heat exchangers. The analyses done during these preliminary design and study phases would be used as input for the individual component development programs.

It is estimated that the preliminary system studies would take about 6 months, and would conclude with the definition of the plant cycle and working fluid, and the development of preliminary plant statepoints and a plant design concept. Following this, the preliminary design of the prototype engine would commence. This would primarily include a detailed mechanical design and assembly layout of the entire CCCBS plant, together with the necessary auxiliaries. Unless the individual component development programs dictate the need for a design change, this preliminary concept would be used as the basis for the fabrication of the prototype engines. The definition of this preliminary plant design is estimated to take about 6 months.

4.1.2 COMPONENT DESIGN AND TESTING

Based upon the preliminary system design studies described above, the detailed design of the individual components would begin at this time. For many of them, scaled-down versions of the components (scaled-down meaning sections of the full-size components, such as 4 compressor stages rather than the full unit, full length sections of the heat exchangers rather than complete components, etc.) would be designed and built, and then subjected to a comprehensive testing program to evaluate the adequacy of their design. For some components, special facilities would be required to conduct these tests. The results from these testing programs would then be used to either modify or substantiate the projected design of the components for the prototype plant.

More detailed descriptions of the component development programs are given in Section 4.2. A summary of the schedules for each of the components are given on Figure 4-1.

4.1.3 MATERIALS

In conjunction with the development of the turbomachinery and heat exchangers, research and testing is needed on many of the alloys required for the high-temperature portions of the plant. This testing would have to be done prior to the detailed design and fabrication of the individual plant components in order to insure that these materials can survive the expected plant conditions.

As described in Sections 4.3 and 9.0, testing is needed on a number of materials that are possible candidates for the construction of various components of the CCCBS. Examples of these components are:

- Turbine blades
- Ducting and piping exposed to temperatures approaching the heat source exit temperature.
- Bearing contact surfaces

It is felt that the major portion of the testing and validation on the candidate materials for these components could be done in time for the detailed design to proceed without any setbacks occurring in the overall development schedule. The major portion of the materials development effort would be expended in the validation of a material for use in the heat exchanger needed for a fossil fired heat source. This component would be subject to the highest temperatures in the whole plant, and would require the use of ceramic materials for the heat exchanger. Based upon the data presently available, however, it would appear that a suitable unit could be built in time for the overall plant integration and the engine qualification tests to be run approximately 5½ years into the program. It does not appear, therefore, that the necessary materials research and development would hold up the overall CCCBS design schedule.

4.1.4 PLANT AUXILIARIES

Development work would also be needed on the various plant auxiliaries in order for them to be ready for the prototype engine tests. These auxiliaries would include a gas bearing helium supply for startup and shutdown, precooler and intercooler heat rejection system, and the plant control system. All of these components would not be needed until the two development engines are ready for testing. Therefore, a minimum of a 3 year period is available between the completion of the preliminary design and the start of the prototype engine tests. This time is adequate to complete the design of the plant auxiliaries, since they would be of a more straight forward design than the turbomachinery or heat exchangers. Some of the development work on these components could even begin during the preliminary study phase, thus allowing another year to be added to the schedule without impacting the completion of the other tasks.

4.1.5 DEVELOPMENT ENGINES

Concurrently with the individual component design and testing, the design of two full size development engines would be in progress. Following the construction of these engines, they would be tested in a closed-cycle helium facility, including a load dynamometer and a water loop for the precooler and intercooler. Data obtained from these tests would primarily be used to verify the performance of the total plant, and to correct any design defects prior to the construction of the refined qualification or production plants.

The need for two engines is to allow testing to proceed without unnecessarily endangering the whole program schedule in case of an engine fault or failure. With two engines, one can be used for long-term normal operation, while the second one could be used to check the design adequacy during a number of expected transients. The testing of the two units could be staggered such that the test data from one plant could be used to develop design modifications for the other plant. This would enable verification of the modifications to be made without unnecessarily delaying the construction of the qualification engines.

The schedule for the development engine phase of the program is shown in Figure 4-1. It is estimated that the initial design phase would take approximately

18 months. Construction of the first engine would take approximately 9 months, while the second engine, due to added experience, would take approximately 6 months. Following this, the engine testing would commence. The first engine would be subjected to approximately a 3 month test, following which it would be removed, disassembled, and an assessment made of the plant performance. Design modifications deemed necessary based on the performance of both engines would be made at this time, and the plant would then be reassembled and readied for additional testing. This redesign and assembly of the first engine would take about 6 months. Following this, the engine would be subjected to an additional 6 month test, primarily to check its survivability during long-term operation.

After the first engine is removed from the testing facility for analysis and redesign, the second engine would be installed and subjected to a 6 month battery of tests. These tests would be more severe than the ones to which the first engine was subjected to, and would include a number of expected plant transients and upset events. Following the successful testing of both engines, the two qualification engines would be built.

4.1.6 QUALIFICATION ENGINES

During the time when the construction of the development engines is in progress, the detailed design of the two qualification engines would commence. This design process would continue through the major portion of the development engine testing phase, and would take approximately 18 months. The major portion of the design effort would be in the modification and refinement of the development engines. The component configuration and design methods would already have been proved during the component and engine development phases.

The fabrication of the two qualification engines is estimated to take about 6 months. This is shorter than the fabrication time for the two development engines, primarily due to the added experience obtained by this period. Both of these engines would then be individually run in a closed-loop helium facility.

The first engine would be run through a series of tests to verify the normal operating performance of the unit. This would include normal steady-state operation at various power levels, as well as normally expected transients such as throttle rampups and rampdowns. Following the successful completion of this test, the second engine would be used to run through the Navy qualification tests necessary to insure that the unit would satisfy the design requirements set down at the start of the program.

One of the qualification tests necessary for naval propulsion systems is the ability to survive shock loadings due to underwater explosions (specification MIL-S-901C). The ability of the CCCBS to survive these loadings would be demonstrated by mounting the first engine (following the initial performance tests) onto a barge and subjecting the entire assembly to a number of underwater explosions.

The testing phase for the first production engine would run about 9 months. Following this, the engine would be removed from the test facility and readied for the shock loading qualification on the barge. The second engine would then be mounted in the facility and subjected to the sequence of testing required to insure compliance with the naval performance and survivability requirements. This qualification test would take approximately 1 year to entirely complete, primarily due to the need to prove that the engine can survive for the expected lifetime of the plant (10,000 EFPH).

4.1.7 ENERGY SOURCE

One component development that has not been dealt with is the energy source. While a detailed analysis of this component is outside the scope of this study, it should be noted that the energy source heat exchanger could impact the turbomachinery development program. Therefore, an estimate of the energy source development schedule is necessary.

In order to conduct the tests on the development engines, an energy source would be needed. Since these engines are used primarily to prove the validity of the overall design, they would not necessarily have to operate at the full

design turbine inlet temperature. This fact would enhance the possibility of a heat exchanger being developed in time for the tests to be conducted.

Based upon the Marine Closed Gas Turbine study performed for the Department of Commerce (Reference 8), it is estimated that a metallic tube heat source heat exchanger could be designed within 3½ years to allow for engine tests with turbine inlet temperatures at approximately 815°C (1500°F). This heat exchanger would enable the engine performance to be verified and still allow the design to be assessed for problems when operating between 871 and 925° (1600 to 1700°F). In order to operate in this range of turbine inlet temperatures, a ceramic tube heat exchanger would have to be developed and built. Based upon the Reference 8 study, it appears as if this unit could be designed in time for the qualification engine tests at about 5½ years into the program. As a result the fossil fired heat exchanger development program would not retard the CCCBS development.

The use of a nuclear powered heat source would require a lengthening of the development program over that required if a fossil fired heat source is used. For example, a nuclear powered aircraft study (Reference 9), indicates a 15 year development program for design, test and qualification of an aircraft nuclear powerplant. This could probably be shortened for a marine powerplant but the first powerplant tests would not occur until about six to eight years into the program.

From the schedule shown in Figure 4-1, the prototype engines could be ready for testing within 4½ years into the program. Since the prime purpose of these engines is to check out the validity of the design of the CCCBS, a nuclear heat source would not be needed at this time. Tests could be conducted with a metallic tube fossil-fired heat exchanger, and any necessary design modifications could be taken care of at this time. The integration of the reactor and CCCBS could wait until the qualification engines have been built.

The fabrication of the turbomachinery would have to proceeed ahead of the reactor construction. Due to the high pressures necessary, there would not be any practical method to test the reactor until the turbomachinery is built. Therefore,

any major problems found in the turbomachinery design should have been corrected prior to the start of the tests using a reactor heat source. This would restrict most of the plant modifications necessary to primarily just those needed on the reactor. Therefore, it would appear that the design of the nuclear heat source would not retard the development of the CCCBS powerplant package.

4.2 COMPONENTS

4.2.1 TURBOMACHINERY

4.2.1.1 COMPRESSORS

The design of any axial flow compressor is not an exact science. The mathematical model used to describe the flow picture through the machine must be greatly simplified if it is to be solved. Therefore, an initial design is made which must rely heavily on previous experience. The approach to the problem is to assume that each blade element acts independently of the one immediately radially above and below it and that this element will, therefore, perform as it did when tested in a two-dimensional cascade. The required performance of each element is determined from the fact that the fluid must be in radial equilibrium. Further assumptions are made as to the effects of end walls, viscosity, shocks, clearance between moving and stationary parts, and growth of boundary layers.

The above assumptions results in enough uncertainties to require test determination and verification of performance. In some instances a new design is not unlike a previous one and requires no further verification. In other cases the state-of-the-art is extrapolated so that testing is required. Such is the case of the CCCBS compressors.

The use of helium as the working fluid requires a large amount of work of compression, resulting in a great number of stages. To keep the number of stages within reason it is necessary to ue larger hub to tip diameter ratios and higher work stages than used in present day commercial practice. The primary aim of the test program is to verify this extrapolation and obtain experimental verification of the aerodynamic adequacy of the design. In addition the test program can be designed to allow optimization of some of the geometry of the compressor and the determination of the effects of operating parameters such as clearances at blade tips and stator seals.

As a part of the CCCBS compressor development program, a four stage full scale model of the intermediate stages of the high pressure compressor and data generated from which design information can be obtained. This information will then be used to design prototype compressors for the CCCBS turbomachinery.

Objectives of Four Stage Model Tests

The overall objective of the program is to obtain information from which compressors can be designed which would require slight or no further modification to produce acceptable performance. To meet this requirement, specific objectives will be established as follows:

- Determine the performance of the original or base point design.
- Determine the optimum number of blades or the solidity of a stage.
- Determine the effects of the axial position or the location of a particular stage on that stage's performance. That is; does a stage perform differently because it follows a large number of stages, and if it does, what is the approximate effect?
- Determine the effect of different blade angle settings on the performance of a stage. (The stage work increases with a decrease in blade setting angle.)
- Determine the effects of shroud seal leakage and rotor tip clearance on stage performance.

Test Facility

In aerodynamic testing a model will simulate a prototype if the flow fields and forces in these fields are similar. In order to achieve exact similarity it is necessary to duplicate in the model the same Mach number, Reynolds number, and flow angles as would be encountered in the prototype. It is generally impractical to make an exact simulation since this exactness can only be obtained in most cases if the two are identical the model therefore becomes the prototype. However, experience has shown that, if the Mach number is low, the effects of Mach number are negligible and that if Reynolds number is above a certain value its effects are small. The use of helium as the working fluid in the cycle causes the Mach number to be low and the power output from the plant makes the Reynolds number high. The major requirement of a test facility for the CCCBS compressor model is, therefore, to be able to simulate a low Mach number and a high Reynolds number. These requirements can be met by testing the full scale model in air in the proposed variable density test rig.

Figure 4-2 is a schematic of the proposed test circuit. The pressure level of the circuit would be controllable between approximately 5 psia and 90 psia by changing the inventory of fluid in the circuit. For the CCCBS tests the fluid used in the circuit will be air, although if conditions should warrant it, other fluids could be admitted. As shown by Figure 4-2 the fluid enters the test section from the plenum chamber, passes through the model and through the adjustable louver valve into the cooler. The louver valve may be adjusted to change the resistance of the circuit and the cooler removes the energy supplied by the 120 HP electric cradle dynometer. The fluid passes through a series of elbows and piping to the upper leg where flow straighteners and screens smooth out the velocity profile before the flow is measured by an ASME long radius nozzle. The diameter of the tunnel piping is approximately 36 inches and the throat of the nozzle is 12.5 inches. The air then passes into the plenum chamber where screens and plates smooth out any flow disturbances before it enters the test section. The plenum chamber is approximately 5 feet in diameter.

A longitudinal section of the proposed four stage compressor model is shown by Figure 4-3. This model consists of an inlet guide vane assembly and stages 9 through 12 of the high pressure compressor.

The test will be controlled from a control panel provided with manometers to indicate pressures measured during a test, and a multi-point temperature indicator. The speed of the compressor model is adjusted by changing the field current of the dynamometer and the load or pressure rise across the model is adjusted by the setting of the louver valve controller.

Static pressures will be sampled by wall taps 0.03 inches in diameter chambered at 45 degrees to 0.06 inches diameter. Final pressures will be sampled by multiple point finger rakes. These rakes will be made insensitive to the direction of flow by the use of suitable shields attached to them. Traverses will be made at the first stage inlet and the fourth stage discharge.

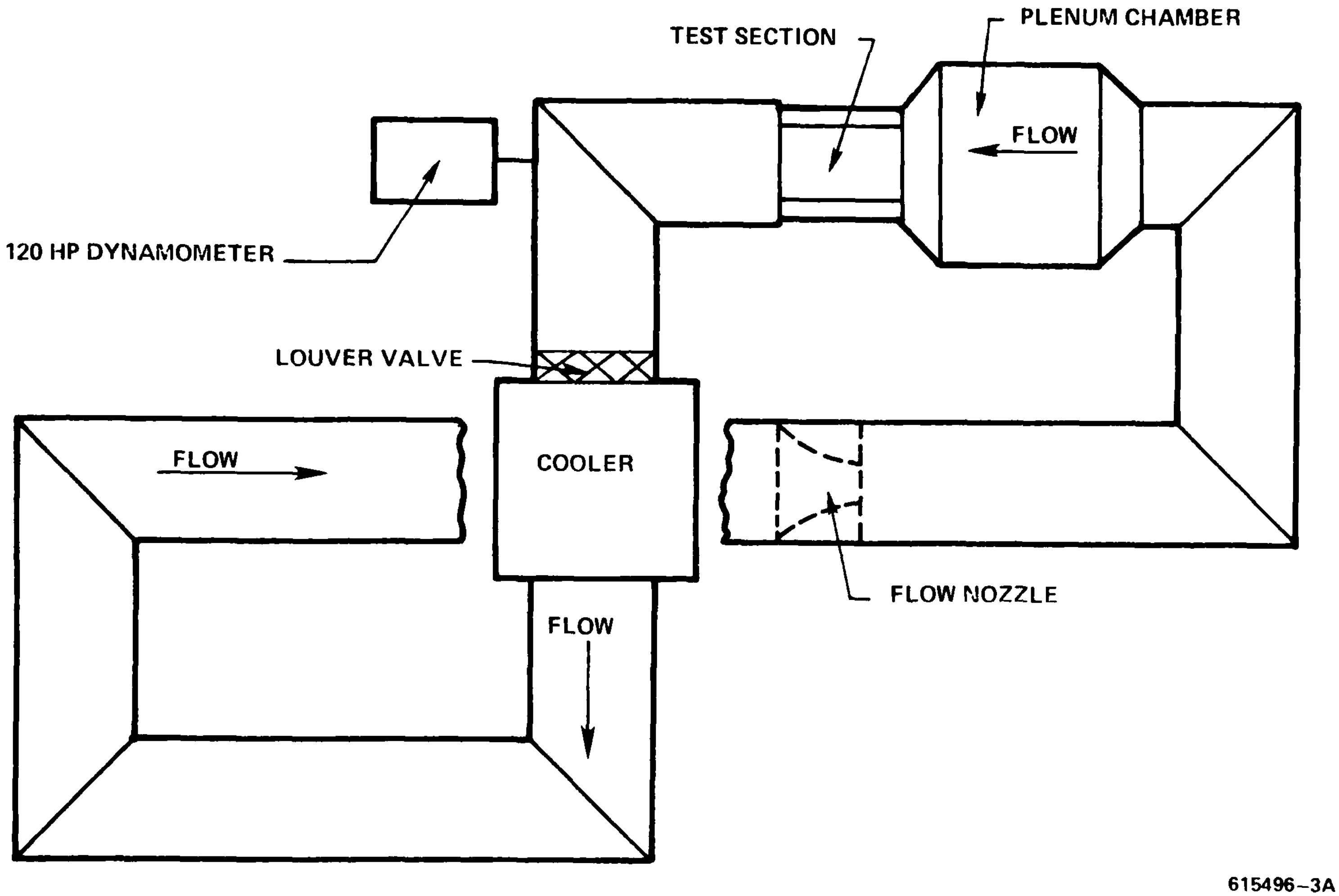


Figure 4-2. Variable Density Test Circuit

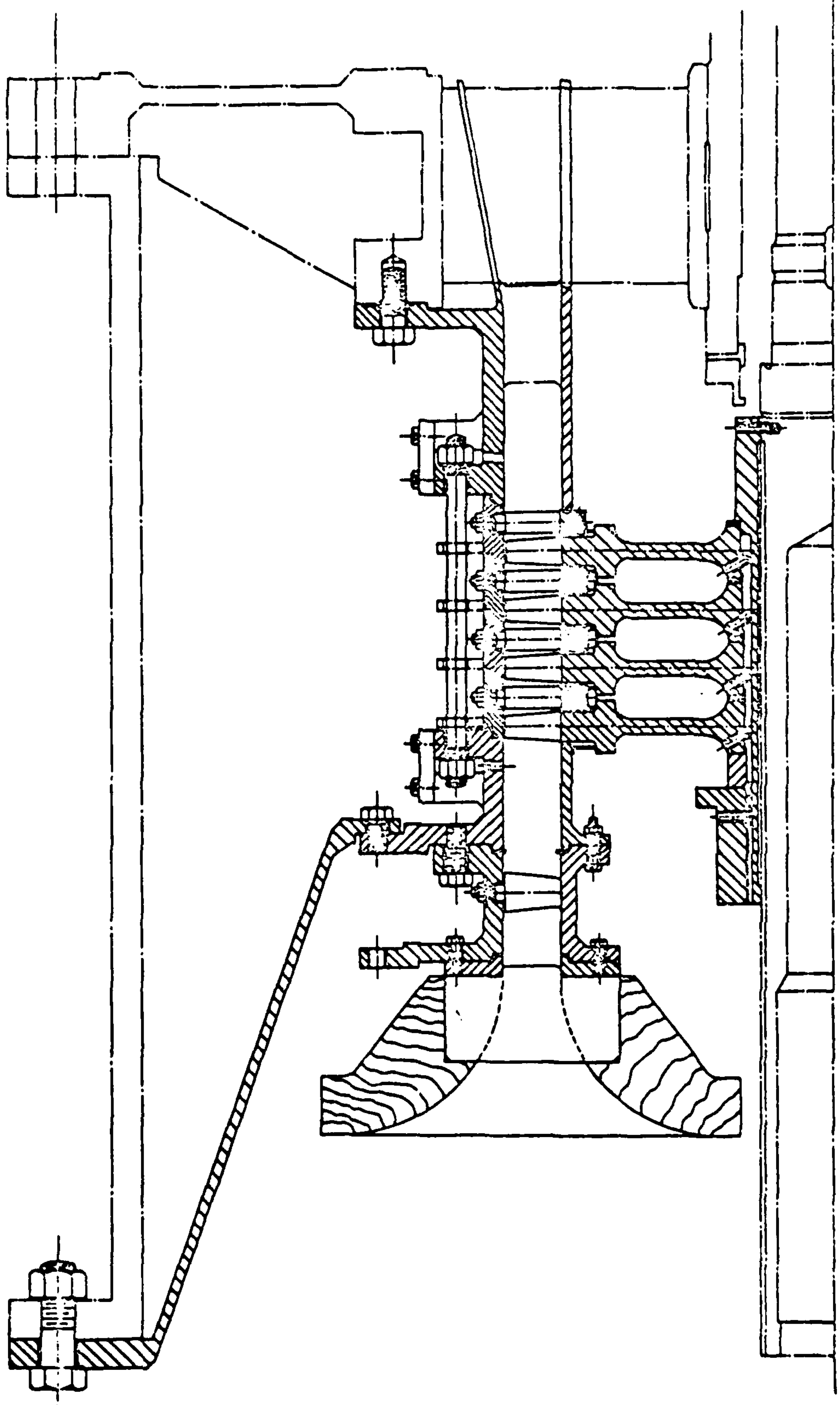


Figure 4-3. Four Stage Compressor Model

Test Program - Four Compressor Stage Model

The first tests run on the model will be performed to check the facility and to determine if the inlet guide vanes are properly directing the flow into the first stage. It is also necessary to determine the pressure level at which the remaining tests should be run to prevent Reynolds number effects from clouding the results.

The facility and instrumentation checks consist mainly of leak detection, calibration of bearing and windage losses, manometer fluid gravity checks and thermocouple calibrations.

Tests will be run to determine the loss coefficient of the inlet bell mouth-inlet guide vane assembly, and to determine the effect of the inlet guide vane stagger angles on the gas flow turning angles. Three stagger angle settings will be evaluated. Traverses will be made on three appproximately equally spaced circumferential stations, downstream from the outlet of the blades. Several radial traverses, approximately 1/8 blade pitch apart, will be made and approximately ten points will be read for each radial traverse. The data will then be reduced and flow weighted values of flow angle, velocity, and pressure obtained.

It has been well established that the performance of turbomachinery is affected by Reynolds number. As this number increases, the effects of viscosity decrease and the effects of inertia forces increase. The performance of an axial flow compressor will, at first, improve rapidly with increasing Reynolds number, and after the "critical" Reynolds number is passed, improve more slowly. The critical Reynolds number is usually between 1×10^5 and 2×10^5 (Reference 1). The Reynolds number of the test model will be variable up to a maximum value of 5×10^5. Tests must be run above the critical Reynolds number, where the effects are small and will not cloud the results. The initial model configuration will be tested at tunnel pressure levels of 1, 2, 3 and 5 atmospheres to determine the effect of Reynolds number of efficiency.

All stages of a compressor operate in a flow field affected by upstream blade rows. The flow disturbances which propagate downstream are generally in the

form of skewed velocity and energy profiles, blade wakes, and wall boundary layers. While it has been established that diffusers and cascades do not reach a condition where the boundary layers stabilize (see References 2, 3, and 4) it is believed that after a few stages, the flow pattern through a compressor reaches an equilibrium condition where the performance of subsequent stages is independent of their axial position.

To determine if subsequent stages reach an equilibrium condition and to get an estimate of the effect on performance of a stage operating downstream of the other blade rows, the following technique is proposed:

1. Traverse the discharge of the model and determine the velocity profile at the design pressure coefficient.
2. Duplicate the discharge velocity profile by mounting the proper size spoilers upstream of the first stage
3. Continue 1 and 2 above until the velocity profiles entering and leaving the model are similar.

The solidity or chord-to-pitch ratio is determined in the initial design of a compressor from two-dimensional cascade tests and modified for three-dimensional cascade tests and modified for three-dimensional effects from the experience of other successful designs. For a given duty, there is an optimum solidity, since the losses associated with viscous drag increase with solidity and the losses due to separation decrease. Tests will be run to determine the optimum solidity for the CCCBS by changing the number of rotating and stationary blades in each stage. Each of the solidity configurations will be tested with and without spoilers.

The maximum efficiency of each configuration tested and the pressure coefficient at which that maximum efficiency occurs will be determined as a function of "stage solidity." Stage solidity is defined as the average of the chord-to-pitch ratio of the stage as shown below:

$$\sigma STG = \frac{\sigma R + \sigma S}{2}$$

This assumes that the effects of rotor and stator solidity are linear and equal and may therefore be combined.

The thermodynamic properties of helium require that a large amount of mechanical work be done upon it to raise its pressure. The work done by each compressor stage is a function of the blade speed and the pressure coefficient. Consequently, by increasing the pressure coefficient, while maintaining the same flow coefficient and speed, the number of stages can be reduced. However, the pressure coefficient cannot be arbitrarily increased without reviewing the effect on overall cycle efficiency of such a change. Where cycle efficiency is important the pressure coefficient must be chosen such that overall performance characteristics are optimized.

Tests will be run in which the blade angles are set to obtain design flow coefficient at higher and lower values of the pressure coefficient. The effects of these changes on efficiency will be evaluated.

The pressure rise across the stationary blading of the CCCBS compressors is approximately 50 percent of the pressure rise across the stage. The stationary blading is supported at the hub by a stationary shroud band which carries metal honeycomb or some other abradable material in close proximity to two knife edged radial seals on the disc rims. These seals span most of the radial gap between the disc and stationary part. While a certain amount of running clearance is necessary to allow for uneven expansions, misalignment, vibration, and the like, it may be possible to improve the seal system if the added expense and complication can be justified.

Since the hub-tip ratio of the compressor is high the seal clearance effects are expected to be significant. The seal clearance will be increased in two steps and test data taken both with and without spoilers, using the base point model configuration. The effect on performance of increased seal clearance for a two throttling seal system on efficiency and the effect on pressure coefficient at maximum efficiency will also be determined. It is expected that the test results will indicate that the seal clearance has a major effect on the performance of the compressor.

The clearance between the tip of the rotor and the outer wall must be large enough to prevent contact when the rotor is spinning, just as was the case of

the disc and the stator seal. This clearance allows fluid to pass from the pressure side to the suction side of the blade, resulting in poorer performance. As the clearance is increased the fall off in performance becomes more pronounced.

The compressor model will be tested at two additional values of rotor tip clearance and the effect of tip clearance on efficiency will be evaluated.

Prototype Compressor Development

The information generated in the four stage model test program will be incorporated in the design of the prototype compressors and will establish confidence in the design methods and criteria used. In model testing, however, there is always some degree of doubt as to how well the actual prototype conditions have been simulated. The history of compressor development contains many instances of stage matching problems which have had to be rectified if all stages were to operate at their optimum level. Final matching may have to be accomplished, therefore, by adjustments to individual stages of the prototype compressors. This final adjustment can be performed during the overall power plant testing in helium discussed in Section 4.1. The information obtained from the four stage model tests should result in prototype compressor designs adequate for the initial powerplant builds will therefore by completely instrumented to allow the stage matching adequacy to be assessed. In this way, it should be possible to dispense with intermediate development steps between the four stage model tests and the overall powerplant tests in helium, therefore, the procurement of prototype compressors specifically for testing in air is believed to be unnecessary.

4.2.1.2 TURBINES

General

The design of an axial flow involves many assumptions of the type mentioned above for the compressor design. As discussed, the CCCBS plant uses helium as its working fluid which makes the turbine design unique in two respects in comparison with aviation and commercial gas turbine experience.

The helium working fluid, coupled with high cycle pressures, impose mechanical limitations on the design which require an extrapolation of the state-of-the-art to lower aspect ratio and higher hub to tip diameter ratio designs than used in present day commercial and avaiation practice. Since both of these effects are generally considered to be detrimental to turbine efficiency, turbine model tests will be run directed at achieving a turbine blade path efficiency goal of 90 percent.

Although it is recognized that the only way to insure the efficiency level of a multi-stage turbine is to test an exact scale model of the multi-stage machine in helium, this approach is not considered to be appropriate for the following reasons:

- Testing in helium is not necessary, since simulated operation in air will give the performance characteristics necessary for a helium design. Therefore, the additional expense of testing in helium could not be justified.
- The cost of blading for a multi-stage model is considered prohibitive, based on the funding judged to be reasonable for turbomachinery development.

Experience with multi-stage steam and gas turbine design indicates that the problems in multi-staging turbines are much less critical than in compressors. This is primarily due to the fact that a turbine accelerates the fluid rather than diffusing it, as is the case in compressors. The accelerating flow results in considerably less severe boundary layer growth on the end walls and the flow pattern is stabilized in one or two stages.

With this background in mind, a two stage model was chosen for test. This approach has the following advantages:

- Two stages represent the point of diminishing returns for a model test, in terms of expenditure for hardware vs design information obtained. Since the first stage sees more nearly ideal inlet conditions, a second stage is required to evaluate multi-staging effects. The second stage receives a flow pattern from the first stage that is more nearly representative of the flow pattern seen by additional downstream stages in a multi-stage machine. Since the growth of the end wall boundary layer is expected to achieve

equilibrium very quickly, the performance of the second stage should be representative of the performance of downstream stages. It is expected that the efficiency of the second stage will be somewhat less than the efficiency of the first stage.

- A two stage model design for relatively high power output has greater potential for detail flow pattern determination by inter-stage traversing. The high power level also minimizes the effect of instrument errors.

Objectives of Two Stage Model Tests

The objectives of the turbine test program are outlined as follows:

- Determine the performance of the design configuration and evaluate if obvious improvements can be made.
- Obtain comparative performance for shrouded and unshrouded blading to determine if the additional complexity of shrouded blading is warranted for the anticipated gain in efficiency with shrouded blades.
- Determine stage interaction effects that will assist in the design of a multi-stage turbine.
- Investigate the improvement of turbine efficiency through optimization of solidity.
- Examine the effects of changing the stagger angle of the stationary vanes.
- Provide verification data for minor field adjustments that may be required.

Test Facility

Air will be supplied to the turbine test facility by a facility air compressor driven by a steam turbine. Design capability of the compressor will be approximatley 50 lb/sec at a pressure ratio of 3:1. A by-pass valve will be provided at the compressor discharge for the selection of any air flow at the desired test pressure level.

A longitudinal section of the test turbine proposed for the CCCBS two stage model is shown in Figure 4-4. Flow will be measured with an ASME long radius nozzle placed a suitable distance downstream from the test turbine in accordance

with ASME standards. A throttle valve, provided at the turbine discharge, will maintain any desired pressure level, up to three atmospheres, in the turbine test section. It will also allow pressurization of the section before each test to assure that there are no leaks in the pressure measuring instrumentation. Power is absorbed by two eddy current dynamometers. One or two stages may be run as a single shaft configuration loaded by one dynamometer or, in the case of two stages, each stage may be loaded by a separate dynamometer. Torque is measured with a Tate-Emergy hydraulic system and a Baldwin strain gauge system. Speeds are measured on chronotach and electronic counter systems.

Test Program - Two Stage Turbine Model

In general a turbine stage is fully defined by the following variables:

- Work coefficient - ψ
- Flow coefficient - Φ
- Degree of reaction - R
- Radial flow pattern
- Hub to tip diameter ratio - DH/DT (at one point in the machine)
- Volume ratio across the machine (determines the area distribution through the turbine)

Once the configuration has been defined, the following variables can affect the basic blade path efficiency:

- Mach Number and Reynolds Number
- Leakage losses
- Velocity triangles
- Blade section losses
- Secondary and wall losses

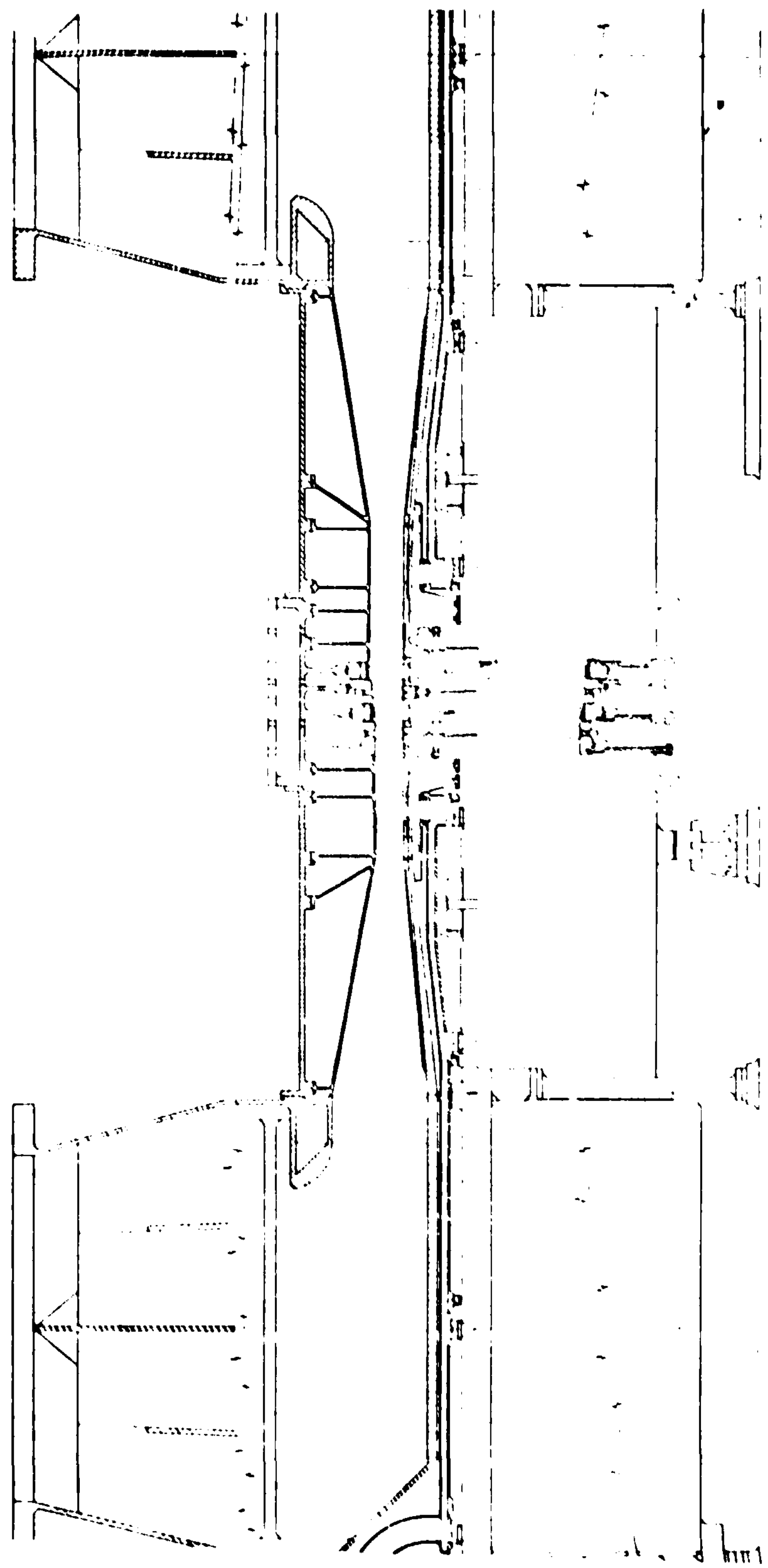


Figure 4-4. Two Stage Turbine Model

In the CCCBS design, Mach Numbers are low and Reynolds Numbers are above the critical value for turbomachinery so neither of these numbers will be a factor in performance variations of the turbine model.

Leakage losses are dependent on the clearance area and the particular seal configuration. In general, they do not cause significant relative variations in the performance field.

For a given loss system, a variation of the velocity triangles causes a variation in efficiency. This is a negligible effect in the region of the design point; however, it is much more pronounced as the distance from the design point is increased.

Blade section losses, secondary losses, and wall losses are closely tied together. Once a configuration has been fixed, the primary reasons for loss changes in a turbine is incidence variation on the blading. Incidence is defined as the difference between the actual velocity vector angle and the design velocity vector angle at a blade row inlet. In general, turbine losses tend to minimize in the region of zero incidence.

A problem of primary importance in any gas turbine cycle is component matching. The referred turbine flow constant will be chosen as a convenient manner for recording flow and pressure ratio characteristics.

It also should be kept in mind that effects of some factors on turbomachinery losses are not known. For example blade wakes cause oscillating incidences on downstream blades and there is evidence from airfoil tests that oscillations can change airfoil drag. This area of nonsteady effects in turbomachinery is virtually unexplored.

Exploratory tests will be run to check out instrumentation and make a preliminary determination of the performance of the second stage of the turbine model operating alone. Traverses will be made behind the stator and rotor to verify velocity triangles, gauging, incidence and the estimate of flow deviation used in the turbine blade design. Deviation is the amount of underturning of the

flow relative to the blade exit angle, and is normally estimated from cascade tests. Cascade tests will probably not be run, since any mismatch of the machine due to an incorrect deviation estimate can be corrected by adjustment of the stator blade angles.

Radial traverses behind the rotors will be made at several referred velocity ratios to determine the effect of incidence on the performance of the rotor.

Initially, circumferential traverses will be performed to obtain representative results of the flow patterns. However, although radial traverses are expected to be satisfactory behind the rotor, agreement between radial and circumferential results behind the stators is not expected to be as consistent. The flow patterns behind the stator usually consist of a series of sharp blade wakes separated by regions of relative flatness. These sharp variations make radial traversing behind the stators a dubious method. Because the flow patterns behind the rotor consist of a series of small undulations, the blade wakes of the stators having been dissipated and those of the rotor having been disturbed by rotation, the error introduced by using radial traverses is small.

Although radial traverses are considered adequate behind the rotor, circumferential traverses will be performed as further corroboration of the radial traverse results. Appropriate stagger angle adjustments can be made, if required, to adjust gauging.

Tests will be run to determine the effect of radial clearance on efficiency and referred turbine flow constant for shrouded and unshrouded rotor blading.

Dynamometer tests will be made to determine the slope of the efficiency line and the referred turbine flow constant line for unshrouded and shrouded rotor blades.

Limiting factors for the required design radial clearance are purely mechanical in nature, and depend on the relative radial growth of the blades and cylinders due to temperature and stress, the axial movement of the rotor

relative to the cylinder, and finally any possible vibratory conditions. To allow a reasonable margin of safety, this clearance should be approximately twice as large for unshrouded rotor blades as for shrouded blades. There are two means of accomplishing the same margin of safety at small radial clearances for unshrouded blades: honeycomb (or other abradable materials) liners in the outer cylinder or "profile" tips.

Honeycomb liners provide a compatible rubbing surface, in the event of rotor blade contact, which permits the installation of closer initial radial clearances than normal.

"Profiling" the blade tips is a means of reducing the rubbing tip surface area of the blade, so in the event of contact, the tips wear and do not cause complete blade failure.

Tests will be run using honeycomb liners to evaluate their effect on performance. Blades with "profile" tips will also be investigated. A test of this configuration will determine if a compromise in efficiency results from reducing the thickness of the blade tip. Judgment, based on past experience, indicates that unshrouded blading can be used in the CCCBS design without sacrifice in performance if a clearance of 0.0017 inches per inch of diameter or less can be maintained. For the CCCBS configuration this seems reasonable. If higher clearances are required, the slight sacrifice in efficiency will probably not justify the additional cost and complication of the shrouded blading.

When a second stage is placed behind a first stage, it is expected that the combination will operate less effectively than each stage separately. A single stage, or a first stage in a multi-stage machine, operates with relatively ideal inlet conditions. Stages behind the first stage are subject to inlet conditions imposed by a previous stage, which are far from ideal. In particular, high incidences are forced on the hub and tip sections of the stationary row, and the stationary row is subjected to nonsteady conditions due to the wakes and vortices shed by the upstream rotor, as well as the weaker wakes carrying through from upstream stators.

In the multi-staging of turbine stages, it is expected that downstream stages will have essentially the characteristic of the second stage and some net improvement in efficiency due to the reheat effect. However, in a two stage test, the reheat effect will be offset by the factors previously mentioned.

To study the stage interaction effect, the first and second stages of the model will be tested separately with essentially ideal inlet conditions to each stage. The two will then be combined to determine performance differences.

In conjunction with this program it is planned that radial and circumferential traverses will be run to help determine the reasons for performance changes and further verify stage interaction effects.

In generaly, the efficiency of the first stage would be expected to be slightly lower than the second, since it has a higher hub to tip diameter ratio, which results in slightly less advantageous leakage and aspect ratio effects. There is no way of predicting the efficiency of the two stages in combination.

Flow constants of the two stages would be expected to be about the same, with the first stage being slightly less than the second, since the end wall boundary layer is a greater percentage of the total passage.

Since the CCCBS turbines will have a large number of stages, the performance of subsequent stages is of particular concern. The premise for the selection of a two stage model was based on the opinion that subsequent stages will perform essentially as the second stage. This option is supported by several observations. No problems have been encountered in matching large numbers of stages, particularly if the volume ratio is low as in the CCCBS turbines. Intermediate stages of steam turbines have achieved efficiencies of better than 90 percent with 12 to 20 stages. This is mainly attributed to the fact that a turbine is basically a fluid accelerating machine and the end wall boundary layers stabilize in one or two stages. Further, the reheat effect works to the advantage of the turbine. Also a turbine is, in general, relatively insensitive to small incidence variations.

Tests will be made on more than one value of stator solidity, to determine its effect on stage interaction and efficiency. It is believed that higher solidity may reduce separation tendencies at high incidences although there would be a net increase in "wetted" surface area.

A series of tests will be run primarily to establish changes in efficiency and flow characteristics with stator stagger angle variation. This information will be required to adjust the cycle matching of the prototype machine at the required inlet temperature. It may be determined that further improvements are possible by using twisted stator blades and increasing rotor solidity. Investigation of these possible improvements will be deferred until shrouded and unshrouded blading and stage interaction effects have been fully evaluated, since these are very basic design considerations.

4.2.1.3 BEARINGS

The achievable characteristics of the turbomachinery bearings can have a significant impact upon the turbomachinery. In addition, both oil and gas lubricated bearings can be considered for the CCCBS turbomachinery. Based upon the analyses done during Year 1, the gas bearing concept was chosen for incorporation in the CCCBS. A description of a representative development program is stated in Section 4.2.1.3.1. However, recognizing the possibility of not being able to develop a gas bearing system in time for power plant testing, an oil bearing development program was generated that would proceed in parallel with the gas bearing program. This is described in Section 4.2.1.3.2.

4.2.1.3.1 GAS BEARINGS

Introduction

The hydrostatic gas bearings proposed to support the various rotating groups defined by Figures 7-1 and 7-4 require development prior to incorporation into expensive turbomachinery. This development is required because hydrostatic gas bearing technology has not been adequately demonstrated on large scale turbomachinery operating on helium gas. Small-scale turbomachinery, such as the 10 kW_e NASA Brayton Rotating Unit (BRU) closed Brayton cycle unit shown in Figure 4-5 have been successfuly operated for over 30,000 hours on individual units. The BRU unit incorporates the type of solid geometry gas bearings with hydrostatic provisions selected for the CCCBS engine, but the bearing journal diameter is 4.445 cm (1.75 in.) compared to the 15.875 cm (6.25 in.) to 25.4 cm (10.0 in.) being considered for the CCCBS engine.

Bearing configurations must be developed to accommodate the thermal gradients, distortions, misalignments, loads and rotor dynamics of large gas turbines while operating with essentially the same gas film thickness as the much smaller bearings. In some cases, the larger bearings may be composed of multiple pads similar in size to the smaller bearings in order to accommodate the mechanical and thermal needs of the large turbomachinery.

In addition, a hydrostatic gas supply system must be developed for large turbomachinery to assure an adequate supply of gas for engine start up, operation, shutdown and emergency conditions.

Development Items

Development is required on three journal bearings and two trust bearings as listed below:

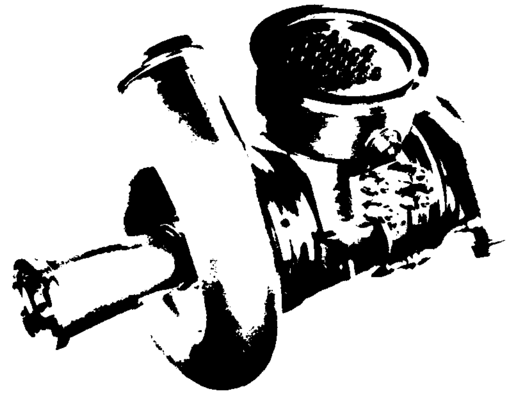
Journal Bearing	Location
15.88 cm dia. (6.25 in. dia.)	Low pressure compressor and high pressure compressor turbine (18,000 rpm)
17.78 cm dia. (7.00 in. dia.)	3600 and 9000 rpm power turbine
25.40 cm dia. (10.00 in. dia.)	3600 rpm power turbine

GARRETT

BRAYTON ROTATING UNIT (BRU)

HIGH RELIABILITY BECAUSE:

- FEW PARTS
- SIMPLE DESIGN
- NO RUBBING CONTACT



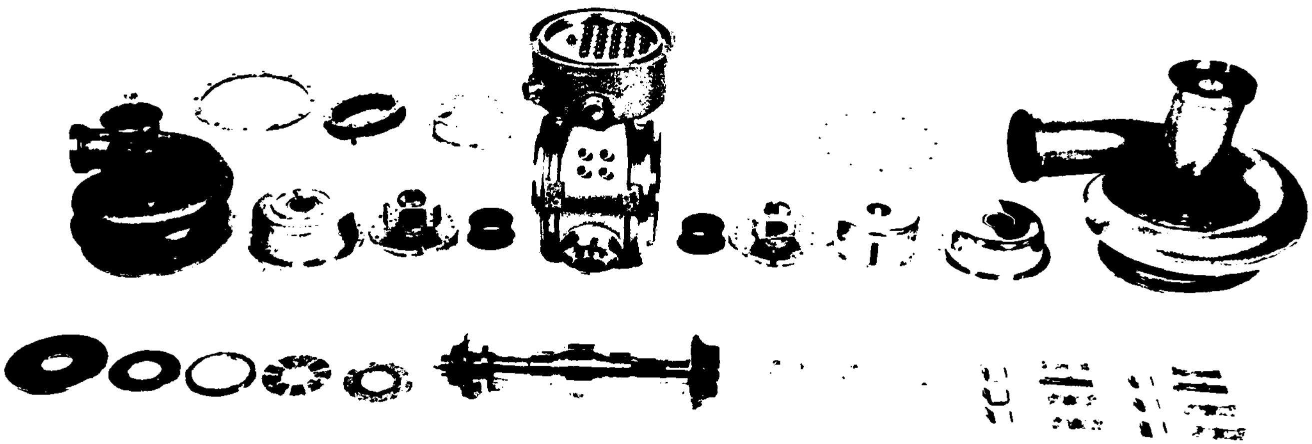


Figure 4-5. Brayton Rotating Unit (BRU)

Thrust Bearing	Location
42.8 KN (10,000 lb.)	Low pressure compressor (18,000 rpm)
59.9 KN (14,000 lb.)	3600 and 9000 rpm power turbine

In conjunction with the bearing development, work is also required on a hydrostatic gas supply system, bearing cavity seals, bearing coatings, and resilient mounts.

Development Program

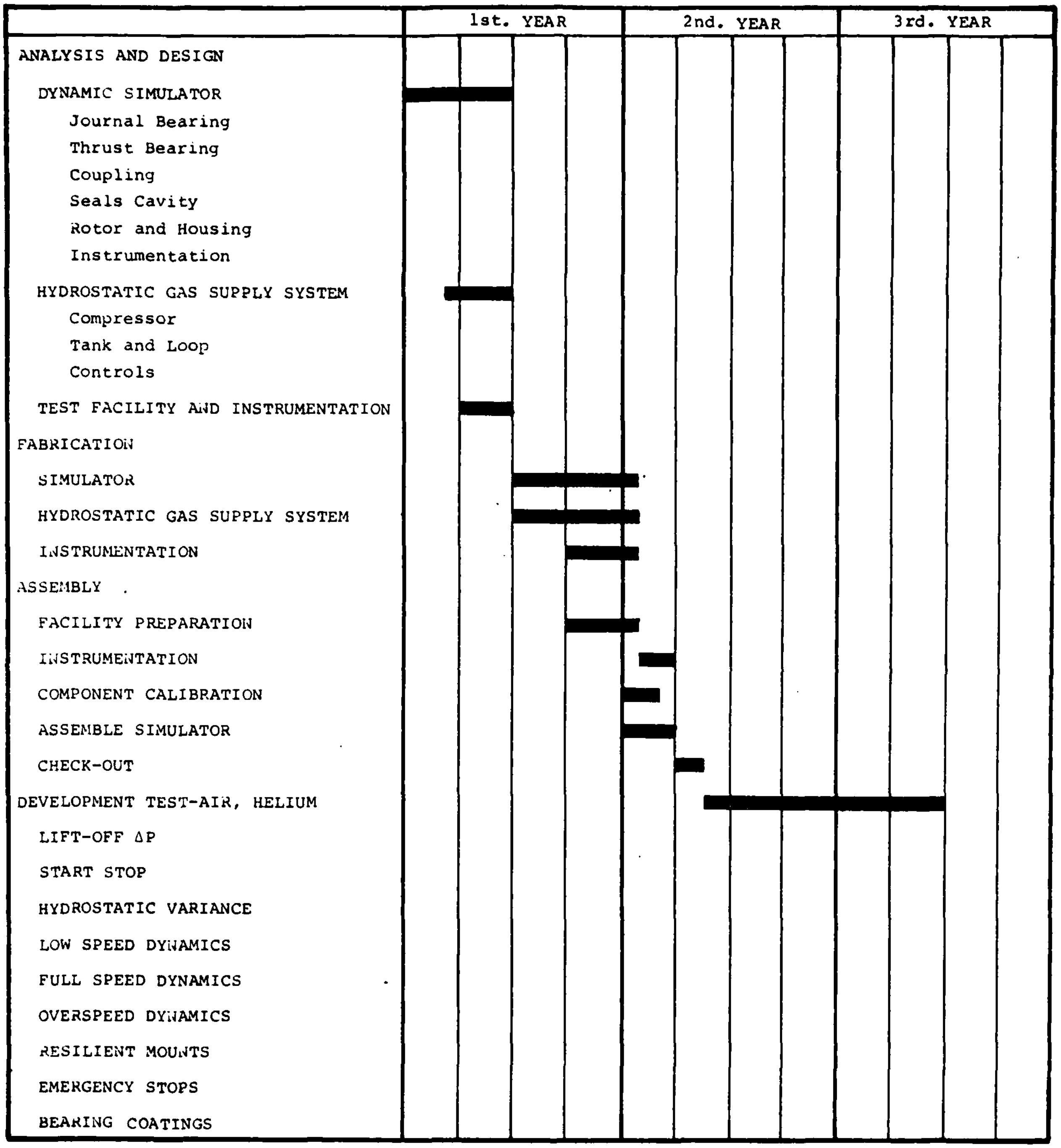
The recommended development program for the CCCBS engine gas bearings and related components is outlined in the schedules shown on Figure 4-6. The details of each task are briefly described in the following paragraphs.

Analysis and Design

A single bearing test rig and a dynamic simulator of the CCCBS engine will be analyzed and designed. The design will provide for development of the various gas bearings, the flexible coupling between the compressor rotating groups and bearing cavity seals. The simulator will incorporate heaters to control bearing ambient temperatures to predicted values. Bearing ambient pressures will be controlled over the specified range. The dynamics of each of the rotors will be monitored by proximity instrumentation.

A complete hydrostatic gas supply system will be analyzed and designed. The system will provide for rotor start up, steady state and transient operation, shutdown and emergency conditions. Controls will be analyzed and designed to provide automatic operation of the bearing system.

The test facility for the single bearing test rig and simulator will be analyzed and designed. Provisions will be made for the drive system, speed control, and instrumentation readout for rotor dynamics, temperatures and pressures. The facility will provide for the integration of the simulator and hydrostatic gas supply system.



DEVELOPMENT PROGRAM SCHEDULE
GAS BEARING SYSTEM FOR
WESTINGHOUSE CCCBS SYSTEM

Figure 4-6. Gas Bearing Development Program Schedule

The analysis and design will require six months to accomplish as shown in Figure 4-6.

Fabrication

The fabrication of the single bearing test rig, simulator, hydrostatic gas supply system, and instrumentation will require seven months to accomplish. The fabrication will be supervised and monitored by Engineering to assure that all components are of the specified material and accuracy.

Assembly

The assembly task includes the preparation of the test facility site, energies, utilities, and installation of site instrumentation. The test components will be inspected and calibrated for contour, gas flow and other parameters. The single bearing testing, simulator and hydrostatic gas supply system will be assembled and functional checks of all components, instrumentation and test facility controls will be made. The assembly task will require 7 1/2 months to accomplish.

Development Testing

Development testing of the single bearing test rig, simulator and hydrostatic gas supply system will require 13 1/2 months to accomplish. Tests will be performed and data analyzed to develop a reliable gas bearing system for the CCCBS engine. Modifications to the bearing designs and other systems will be made as required.

As a minimum, the bearings and related systems will be subjected to the following tests in both air and helium gases at temperatures up to 260°C (500°F).

- Hydrostatic Lift-off - Pressure variations
- Gas film Thickeness
- System Dynamics
- Resilient Mounts and Damping
- Bearing Coatings
- Emergency Stops

Hydrostatic Lift-Off-Pressure Variations

The magnitude of pressure necessary to initially separate the bearing and journal surfaces (lift-off pressure) is expected to be greater than that magnitude to sustain a stable gas film. This is because, prior to lift-off, the effective bearing surface area is largely confined to the pad recess area. The necessary supply pressure for lift-off must necessarily therefore be defined under conditions representative of the static bearing load. This start-up pressure requirement must be an integral criterion for the start-up system design. Parametric optimizations will be required to minimize the difference between lift-off and operating supply pressures, consistent with other bearing performance requirements.

Gas Film Thickness

Confirmation of predicted operating gas film thicknesses is necessary to verify the bearing performance integrity and to identify any undesirable performance penalties. Film thickness measurements will be determined by proximity probes mounted with the bearing pads and directed at the journal. Additional probes will monitor journal and pad motions relative to ground; thereby, comprehensive characterization of the journal bearing system will be quantified. This latter characterization is necessary to identify possible pad-journal dynamic instabilities.

Similar instrumentation will be provided for thrust bearings.

System Dynamics - Start-Stop, Low Speed, Operating Speed, Overspeed

These tests will be fundamentally directed toward:

- Identifying system requirements as dictated by start-stop capability of the rotor system
    - Gas bearing supply requirements
    - Pressure requirements
- Verifying rotor dynamic system stability throuhgout the operating speed spectrum

The gas management system for the gas bearing system will be largely dictated by the start-stop requirements of the gas bearings. Critical parameters are flow rates, total integrated flow, operating time duration between starting and stopping, and maximum required supply pressures. To this end, with the full dynamics system represented, gas bearing operating parameters will be measured for representative start-stop cycles. These will include bearing supply flow rates and pressures and time durations during accelerations and decelerations.

Empirical verification of system dynamic stability is essential. Rotor amplitudes, transient and steady state, and frequencies must be determined for the properly simulated dynamic system. Measurements will be made by proximity probes of rotor motions, bearing pad motions, and rotor-relative-to pad motions.

Resilient Mounts and Damping Tests

The characterization of the gas film bearing system dynamic properties is necessary for accurate prediction of the bearing-rotor system dynamic behavior. A first order concern is to gain as much confidence as is realistically possible that each rotor-bearing system is dynamically stable prior to empirical verification by dynamic simulation testing. If dynamic stability evaluations are exclusively relegated to test rigs, program risks in terms of schedule slippage, hardware costs, and redesign become very high.

Therefore, early determination of bearing performance parameters relevant to dynamic stability analyses is essential. The specific parameters to be determined are journal bearing direct and cross coupled stiffness and damping coefficients. These will be determined by conducted using single bearing test facilities. Stiffness coefficients will be determined by measuring journal displacement vector relationships to applied load vectors. Damping coefficients will be determined from measured journal response to impulse journal load applications. By combining these parameters with rotor dynamic stability studies, the required bearing performance parameters will be defined, evaluated, and verified during the single bearing test rig stages of the program.

Bearing Coatings

The predicted shock loads on the CCCBS engine bearings are sufficiently high that the load capacity of those bearings will be momentarily exceeded and rubbing contact will take place between the bearing pads and shaft journals. It is imperative that the pad and journal materials be compatible and maintain relatively smooth surfaces during the rubbing contact. Bearing coatings of metallic and non metallic materials have been used successfully on smaller gas bearing supported turbomachinery subjected to shocks up to 500 g's. The development of coatings for the CCCBS engine gas bearings is considered to be a major task due to the large energy transfer during the rubbing contact of high inertia rotors systems. Relatively low bearing temperatures are predicted for the CCCBS engine and non metallic coatings such as Teflon S may provide adequate protection during shock loads.

Coating development for the CCCBS bearings would encompass a study of the rubbing energies during the projected shock pattern in conjunction with a materials and process development. Single bearing test rigs would be used initially with selected coatings being evaluated on the simulator. Roll down tests would be conducted by depriving selected bearings of the hydrostatic gas supply at increasingly higher and higher speeds.

Emergency Stops

Emergencies can develop with any power generating equipment that require an immediate and rapid stop of the turbomachinery. Turbomachinery mounted on gas bearings ideally would be supplied with a separate and independently operated hydrostatic gas supply system that would permit a safe roll down of the turbo machinery without damage to the bearings or turbomachinery. In the event of a failure of the hydrostatic gas supply system an emergency backup system should be supplied. In the case of the CCCBS engine, a reserve gas supply tank and controls with suitable engine braking to decrease the roll down time would be acceptable. The development of such a system would be integrated with the controls and operation of the hydrostatic gas supply system required for normal operating and development testing of the simulator.

Reporting

Monthly technical reports and program management reports will be provided for the complete 30 month development program. A final report will be submitted.

Experimental Research

Introduction

A separate research effort not included in the development plan for the CCCBS engine gas bearings does offer enhancement of the specific development program and provides growth of gas bearing technology to general applications extending to utility scale gas turbines with rotor weights in region of 266.9 KN (60,000 lbf). While the proposed development program would be adequate to develop the hydrostatic gas bearings for the CCCBS engine, a parallel research program would support the development program and provide the rigorous study necessary to extend the technology to a broad field of applications.

The following paragraphs define the need for a research program and outline the objectives and approach that could be taken. In some cases, the test rigs and simulators for the development program could be shared with the research program with an adjustment in development schedule.

General Scope - Research Program

The size magnitudes of the externally pressurized bearings and rotor systems required for the 70,000 hp CCCBS engine constitute a major extrapolation beyond the existing state of gas bearing applied technology. While the major design concerns can for the most part, be rather readily anticipated, their resolutions will require combined extensive analytical treatment and empirical verification. An unfortunate reality is that most analytical predictions of load capacity surfaces; journal bearing surfaces are treated as ideally cylindrical, and thrust bearing surfaces are treated as ideally plane in the initial sizing definitions. The bearings have been sized for this application using these idealizations with appropriate conservatism included with the objective of compensating for the non ideal conditions that will result in the actual hardware.

Almost without exception, the actual hardware deviations from the model idealizations have the consequence of compromising bearing performance. Causes for these deviations originate with the manufacturing and assembly processes of the components and system. They are compounded by operational effects such as misalignments and elastic (or plastic) distortions and possibly creep and relaxation of residual stresses introduced in the manufacturing process.

A fundamental purpose of the experimental research is to quantify these effects. It is necessary that they be evaluated under conditions that realistically simulate actual operating conditions. To this end, it is necessary that before the fact predictions be made of anticipated environmental conditions such as temperature distributions, elastic deflections shock loads, misalignments, manufacturing tolerances, and operational variables. These conditions then serve as a foundation for design of the experimental facilities and test plans.

Test Objectives

The research test program will be divided into two elements: (1) single bearing test rigs for the journal and thrust bearings, and (2) rotor-bearing system dynamic simulation.

- Single Bearing Test Rig Objectives
    - Evaluation and verification of bearing load capacity versus film thickness
    - Identification and resolution of pneumatic instabilities
    - Effects of bearing-journal misalignments on load capacity
    - Determination of pneumatic system empirical constants: orifice discharge coefficient, gas film compensation, etc.
    - Measurement of bearing direct and cross coupled stiffness and damping coefficients
    - Effects of thermally and pressure induced elastic distortions
    - Identification and evaluation of bearing hybrid self-acting and externally pressurized effects
    - Bearing surface material wear compatibilities

- Non-isothermal gas film temperature effects
- Bearing film thickness response to varying pulse width impact loadings and load directions

● Integrated Bearing - Rotor System Dynamic Simulator Objectives
  - Identification and resolution of pneumatic instabilities
  - Evaluation and verification of rotor system amplitude response to imbalance
  - System stability as a function of rotor imbalance
  - Bearing dynamic loads versus rotor system imbalance

Test Rigs

● Single Bearing Test Rig Capabilities - The journal and thrust bearing test rigs will be designed to represent prototype design configurations for the intended system. Capability will be provided to simulate expected environmental conditions such as heat fluxes, ambient temperatures and pressures, and externally supplied flow conditions, force loadings, both static and dynamic, and misalignments.

  Instrumentation will be provided to measure elastic distortions, operating film thicknesses, component temperatures, supply gas flow rates and temperatures, journal displacement vector response to applied load vectors both static and dynamic, gas film pocket pressure, and bearing pad response to journal orbit trajectories.

● Dynamic Simulator - The dummy rotor will be designed to simulate actual mass, inertial, and stiffness properties of a selected system.

  Instrumentation will be provided to monitor rotor dynamic response to imbalance and shock loads.

The parallel research program could be accomplished in approximately thirty-six to forty-two months.

4.2.1.3.2 OIL BEARINGS - BACKUP DEVELOPMENT

While it is believed that gas bearings suitable for the CCCBS can be successfully developed, the development program is expected to extend over several years, during which time the turbine and compressor components will be undergoing rig testing. Ideally, the program should deliver a fully developed gas bearing system suitable for incorporation into the first complete CCCBS powerplant test assembly at the completion of the turbine and compressor component test program. However, recognizing the uncertainties associated with new technology developments and the absolute necessity to ensuring the availability of reliable bearing system in time for powerplant testing, prudence dictates that a backup oil bearing development program be undertaken in parallel with the gas bearing program. The oil bearing system must be interchangeable with the gas bearing system with a minimum perturbation of the turbomachinery rotor and stator geometry, so that either system can be readily incorporated into the powerplant.

At a suitable point in the initial component development phase, a decision will be made to incorporate either the gas bearing system or the backup oil system into the first complete powerplant test assembly. The decision will obviously depend on the development status achieved in the gas bearing system and its predicted reliability in the powerplant assembly. In the event that the decision is made to incorporate the oil bearing system into the final powerplant build, the development of the gas bearing system will continue in parallel for eventual incorporation into later powerplant development builds. On the other hand, if the decision should be made to incorporate the gas bearing system into the first powerplant build, the oil bearing system would be available as a backup and could be quickly substituted if the initial reliability of the gas bearing system in the powerplant should prove deficient.

The research and development program must therefore provide for oil lubrication and seal system development necessary to provide an adequate interim system early in the program. Since the oil bearings would be intended for eventual replacement by gas bearings, the oil seal performance probably need not be developed to the high standards which would be required for a permanent installation. However, the achievement of adequate performance to permit

turbomachinery development to proceed early in the program, unhampered by lubrication and sealing problems, will necessitate that certain minimum provisions be made in the design and development of the system.

CCCBS Powerplant Oil Lubrication System

The CCCBS powerplant lubrication and sealing system may be considered as four basic subsystems:

1. A conventional turbine-type lubrication loop is used to supply oil to the thrust bearings.

2. A closed circuit is used to supply oil for the turbomachinery journal bearings. Oil leaking toward the gas, or cycle side of the bearing housing, is opposed by a helium gas barrier. This oil becomes mixed with helium, and is, therefore removed for purification.

3. The third subsystem is a cleanup or separation system designed to recover the helium and and the oil for reuse in the plant.

4. The fourth subsystem provides a method of sealing the working fluid within the plant without use of the main oil pumps.

In the closed lubricant circuit, the oil is pumped from a closed reservoir, through a filter, a cooler, and a regulator to the bearings. From the bearings the oil is orificed back to the reservoir. Two floating ring seals, one on each side of the bearing, restrict the side leakage of oil. Tilting pad bearings are used since loads are light, approximately 0.7 MPa (100 psi) and plain journal bearings would be prone to oil whip at high speeds.

A small amount of cycle gas is bled from the compressor discharge and passed through a filter and cold trap to remove contaminants. The clean gas is then introduced between two labyrinth seals, at a pressure slightly greater than the adjacent cycle fluid. A barrier is thus formed which prevents contamination of the cycle helium by the lube oil, and also prevents hot cycle gas from contacting the bearing. Some of this clean barrier gas leaks toward the

bearing where it mixes with lubricating oil and is scabenged to a drain tank. The fluid oil settles out and a float valve vents the liquid back to the lub oil reservoir. The gas, some oil droplets, and oil vapor flow into a coalescer or separator. Gas which is separated then passes through an orifice or regulator which holds the drain tank at a pressure slightly below the barrier gas. An absorbant filter finishes the cleanup process and the gas returns to the cycle at the precooler inlet.

A method of sealing the working fluid to the plant was discussed in Section 7.1 and is illustrated in Figure 7-4 as applied to the 3600 RPM power turbine output shaft. A pneumatically energized piston is used to engage a static O-ring seal at shutdown. During a normal operation, with the gas bearing system, the pressure boundary is established by a double face seal using water at high pressure between the seals. This arrangement could utilize oil instead of water in the initial development phase and could be integrated into the oil bearing system. Inboard oil leakage in this case would be scavenged to the drain tank together with drainage from the other turbomachinery bearings. Outboard leakage would drain by gravity to an ambient pressure tank from which it would be returned by means of a transfer pump to the lub oil reservoir. The oil can be vacuum treated at this point if necessary to prevent air from entering the cycle. Since some helium will come out of solution in the reservoir a small compressor can be provided to pump the released gas back to the drain tank thereby maintaining reservoir pressure.

In the overall system the turbomachinery has six pad type journal bearings and one shaft end to be sealed. The bearings fall into two groups according to the local cycle pressure level. The two groups are:

1. L.P. Compressor Inlet Bearing
   Power Turbine Outlet Bearing

2. L.P. Compressor Outlet Bearing
   H.P. Compressor Inlet Bearing
   H.P. Turbine Outlet Bearing
   Power Turbine Inlet Bearing

The bearings in the first group are both exposed to the lowest cycle pressure region of 3.1 MPa (450 psia) to 3.4 MPa (500 psia) at full power. The bearings in the second group are all exposed to the intermediate cycle pressure region of 5.8 MPa (850 psia) to 6.1 MPa (890 psia). As a result, at least two drain tanks will be required, one for the first and the other for the second group of bearings. However, it may be necessary to provide a drain tank for each of the four bearing housings to avoid excessive inter-bearing housing gas flow and resulting seal leakage under transient operating conditions. In this case the six bearings would be grouped into the four housings as follows:

- Housing #1 - L.P. Compressor Inlet Bearing
- Housing #2 - L.P. Compressor Outlet Bearing
H.P. Compressor Inlet Bearing
- Housing #3 - H.P. Turbine Outlet Bearing
Power Turbine Inlet Bearing
- Housing #4 - Power Turbine Outlet Bearing

Oil Lubrication Development and Test System

The development and test system will reporduce the essential features of the CCCBS powerplant backup oil lubrication system. A representative bearing, shaft and seal system will be simulated in a motor driven test rig. The journal bearings will probably be of proprietary manufacturer such as Orion, in which case they will incorporate floating seal rings to permit flooded operation without excessive oil discharge towards the seals.

The test rig will incorporate the necessary labyrinth seals to control the helium buffer gas which enters the bearing housing and oil system. The rig will also be capable of accommodating a pressure boundary gas seal utilizing a tandem seal arrangement with pressurized liquid (oil or water) between the seals. Seals of this type are manufactured by the Crane Packing Company and Figure 7-4 illustrates such a seal applied to a 3600 RPM power turbine for mechanical drive applications of the CCCBS. Provision will also be made for testing a pneumatically operated static shutdown seal to enable the pumps of

the pressurized liquid pressure boundary seal to be shutdown during periods when the shaft is not rotating.

The lubrication system components illustrated in Figure 4-7 will be installed with the rig and the necessary equipment will be provided to charge the rig with high pressure helium and control the pressure levels during operation.

Tests will be performed to evaluate the operation of the lubrication system, the bearing housing seals and the pressure barrier seal system over the range of pressure and speed conditions to be encountered in the CCCBS powerplant. The tests will be performed to simulate the various levels of buffer gas and drain tank pressure corresponding to the several bearing groups in the CCCBS powerplant.

The test rig and lubrication system components will be fully instrumented with pressure, temperature and flow measuring devices to faciliate these evaluations.

It is anticipated that the development program will include the evaluation of alternative bearing and seal designs and refinements to the system. This work is expected to proceed on a routine basis until the gas bearing system is sufficiently developed to be incorporated into the CCCBS powerplant test assemblies.

4.2.1.4 SEALS

Hydrostatic gas bearings depend on differential pressure to provide the design load capacity. Therefore, each bearing must be supplied with a high pressure source of gas and the bearing cavity must be maintained at a lower pressure so as to assure the correct differential pressure. The bearing cavities of the CCCBS engine are vented to various pressure sources as shown in Figure 4-7, and shaft seals at each bearing cavity are used to minimize leakage to or from adjoining areas at other than the desired vented pressure. The seals must be designed to minimize leakage, control thrust balance, and function over a range of rotor eccentricities that vary seal clearances. The seals became an integral part of the gas bearing development as they affect bearing operation as much as bearing characteristics affect seal design. The development of the seals for the CCCBS engine is therefore, included in the develop-plan for the hydrostatic gas bearings.

Labyrinth seals are considered as the primary solution to the bearing cavity pressure control. However, in the event that labyrinth seals cannot be designed to meet the requirements and characteristics of a gas bearing supported CCCBS engine, gas bearing face seals with minimum leakage, excellent tolerance to raidal shaft displacement, lower power loss, and large axial tolerance would be developed in conjunction with the gas bearings.

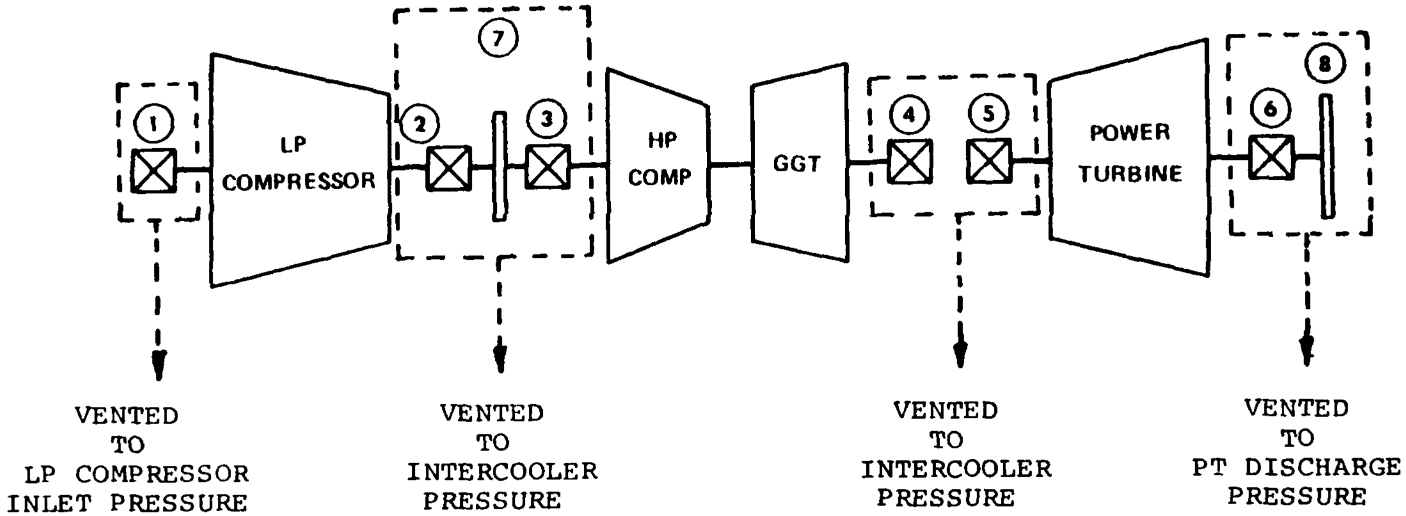


Figure 4-7. Vented Bearing Cavities

4.2.2 HEAT EXCHANGERS

Prior to the construction of the heat exchangers for an actual CCCBS application, a development program is necessary to evaluate the adequacy of their design. This would include investigation of flow distributions, pressure drops, heat transfer coefficients, fabrication, and ability of the unit to survive pressure and thermal transients and those shock loadings generated to satisfy the shock requirements of the MIL-S-901C (Navy). The required development program needed to assure the adequacy of the design would be primarily aimed at the precoolers and intercoolers, due to their larger size complexity. The recuperator can draw, to a large extent, on the data base already developed for gas-to-gas tube and shell heat exchangers.

4.2.2.1 PRECOOLER AND INTERCOOLER

INTRODUCTION

The proposed finned tube precooler and intercooler for the Westinghouse 70,000 hp CCCBS engine are formed into annular shapes in which the fin tubes spiral four full turns. The spirals start at four locations to provide a four pitch spiral of each tube set. The water flows in the circumferential direction through the tubes and the gas flows over the finned outer surface of the tubes in a cross-counter flow arrangement. The proposed designs involve proven and conventional finned tube heat exchanger technology, but the design and packaging is unusual in the following areas and requires development:

- Flow distribution
- High Reynolds number heat transfer and pressure drop
- Tube to manifold joint process

In addition, the conventional heat exchanger structural design, thermal stress analysis, dynamic loading, and acoustical considerations must be integrated with the precooler and cooler developments.

Design for Proper Flow Distribution

The water flow has a tendency to be higher in the shorter tubes (at the inner radius) and lower in the longer tubes (at the outer radius) to equalize the pressure drop. This tendency is opposite from the desired effect, considering the heat transfer requirements, of having the highest flow in the longest tubes and the lowest flow in the shortest tubes. To overcome this tendency, both manifold shaping and tube orificing will be investigated to provide the proper flow distribution for maximum heat transfer.

Heat Transfer and Pressure Drop Design Data

Very little heat transfer and pressure drop data is available at higher Reynolds numbers, for flow over finned tube banks. Data from Russian literature indicates that the normal heat transfer relationships that are used for flow over finned tube banks are not applicable for the higher Reynolds numbers expected in these

designs. It is also anticipated that the normal friction factor relationships would also not be valid at the higher Reynolds numbers expected in these designs. In order to evaluate these high Reynolds numbers effects, a literature survey coupled with testing at high pressure with air is required. A test heat exchanger module would be constructed using the finned tube proposed for the final design. At least ten rows of tubes would be used in the test core to assure fully developed flow over the tube bank. This test core would be operated over the range of Reynolds numbers expected in the CCCBS cooler and intercooler designs. Water cooling would be used inside the tubes and both heat transfer and flow friction would be evaluated on the finned outer surface of the tubes. The heat transfer data would be taken with varying water flows so that a Wilson plot could be constructed. The Wilson plot is a plot of the overall thermal conductance (U) versus ($1/V^{0.8}$) of water. This plot is extrapolated to $1/V^{0.8}$ of 0 to get the finned outer surface thermal conductance.

The finned outer surface thermal conductance as a function of air flow would be reduced to Colburn J-factor data as a function of Reynolds number. The pressure drop of the test core would be reduced to Fanning friction factor data as a function of Reynolds number. This data would be used in a final design of the CCCBS cooler and precooler.

Fabrication Process Development

Fabrication processes would be developed for the tube to manifold joint which could withstand the shock and vibration requirements without leakage. Samples of both brazed and welded tube joints would be subjected to simulated shock and vibration conditions expected in the final design. The samples would be helium leak tested both before and after the simulated loads. A helium leak of less than 10^{-6} cc/sec would be required to pass this test. The samples would also be subjected to proof and burst pressure tests to verify the structural design.

Structural Design

Each proposed heat exchanger will be analyzed to verify capability to withstand stresses imposed by operating and environmental conditions:

1. Pressure Stresses

The criteria of adequacy are:

- There shall be no permanent deformation as a result of the application of a proof pressure P_p = 2.0 x maximum operating pressure

- There shall be no rupture as a result of the application of a burst pressure P_b = 4.0 x maximum operating pressure

If conducting these tests at the prescribed temperature proves impractical the applied test temperatures will be increased by the ratios of yield and ultimate material strengths respectively at room temperature and at the prescribed temperature.

2. Thermal Stresses

Both steady state and transient conditions will be investigated. Particularly during start up, a temperature lag occurs in the heaviest sections of the heat exchanger. Unless the time period for the thermal transient is equal to or longer than the time constant for the heaviest section in the unit, thermal stresses in excess of the steady-state levels will be calculated using temperature distribution maps developed by the Thermodynamics Group.

Creep considerations will not be addressed because of the relatively low operating temperatures.

Fatigue life capability will be calculated using both pressure and thermal stresses, with the operating duty cycle of the units. Specifically, the heat exchangers will be configured such that

$$\sum_{o}^{p} \frac{n_{i\ \mathrm{actual}}}{N_{i\ \mathrm{predicted}}} \leq 1$$

where

i = discrete number of loadings (stress applications)

n_i actual = number of times the particular loading is repeated during the intended operational life

N_i predicted = number of times the particular loading was to be repeated in order to cause failure

The calculation of $N_{i\ predicted}$ will be made through use of AiResearch x 0870 low cycle fatigue computer program. This computerized calculation uses the Wetzel-Morrow method in conjunction with the Neuber hyperbolas to determine the stabilized total cycle strain range. This strain range is then input to the Manson-Hirshberg equation to predict cycle life. Some of the details involved in the method are shown on Figures 4-8 and 4-9.

3. Dynamic Loading Stresses

The more severe dynamic condition that each heat exchanger (precooler and intercooler) has to withstand is likely to be vibration. Shock and acceleration test conditions are addressed after the unit design has been iterated to cater to vibration test requirements.

The heat exchanger attachment interfaces are sized to accommodate the prescribed input G-levels multiplied by an amplification factor of Q = 10. It will be ensured that stress levels associated with this condition are at or below the endurance limit of the material.

Internally to the heat exchangers, the tubes will be supported in a manner such that the individual tube and the tube bundle natural frequencies in a radial and in an axial direction do not coincide with the frequency range within which adjacent components operate. In the proposed precooler and cooler design, the tube supports would be rectangular in shape and placed normal to the axis of the tubes.

4. Acoustics Considerations

A heat exchanger can act as a sound pipe and have acoustics frequencies that are a function of the characteristic length for acoustic vibration. This length is the depth of the heat exchanger air flow passage.

$$f_a = \frac{m\ c}{2\ y}$$

where

fa = acoustic frequency, Hz

m = dimensionless node number

c = velocity of sound in airflow, ft/sec

y = effective flow passage depth, ft

PREDICTION OF UNIT LIFE FROM LEVELS OF APPARENT ELASTIC STRESS REACHED DURING TRANSIENT LOADINGS (AIRESEARCH COMPUTER PROGRAM X0870)

- WETZEL-MORROW ELASTIC-PLASTIC ANALYSIS USING NEUBER HYPERBOLAS TO OBTAIN STABILIZED CYCLIC STRAIN RANGE.

- NUMBER OF CYCLES TO CRACK INITIATION OBTAINED USING MANSON-HIRSCHBERG EQUATION OR MODIFIED MANSON-COFFIN EQUATION

$$\epsilon_{TOT} = 3.5 \frac{\sigma'_f}{EN_f^{0.12}} + \frac{(\epsilon'_f)^{0.6}}{N_f^{0.6}}$$

WHERE σ'_{f_1} = TRUE ULT. FAILURE STRESS } FROM PULL TEST

ϵ'_{f_1} = FAILURE STRAIN } FROM PULL TEST

n = STRAIN HARDENING EXPONENT

- ACCUMULATIVE FATIGUE DAMAGE. (MINER-PALMGREN RULE)

$$\sum_{o}^{p} \frac{\eta_i}{N_i} \leq 1$$

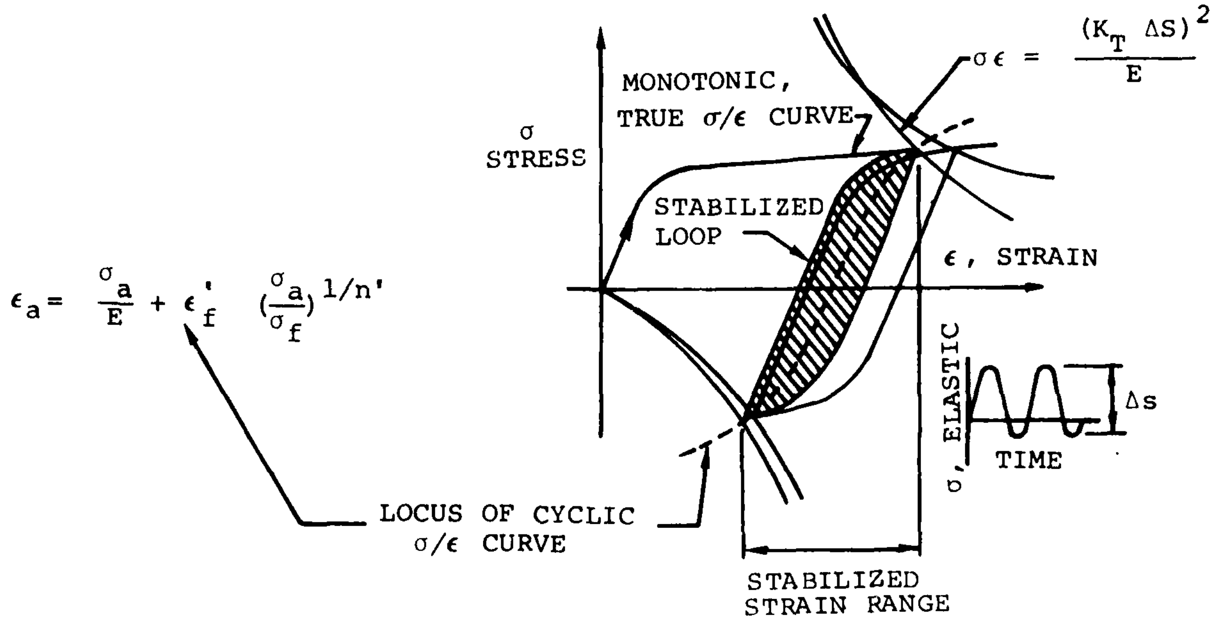


Figure 4-8. Fatigue Life Calculation Method

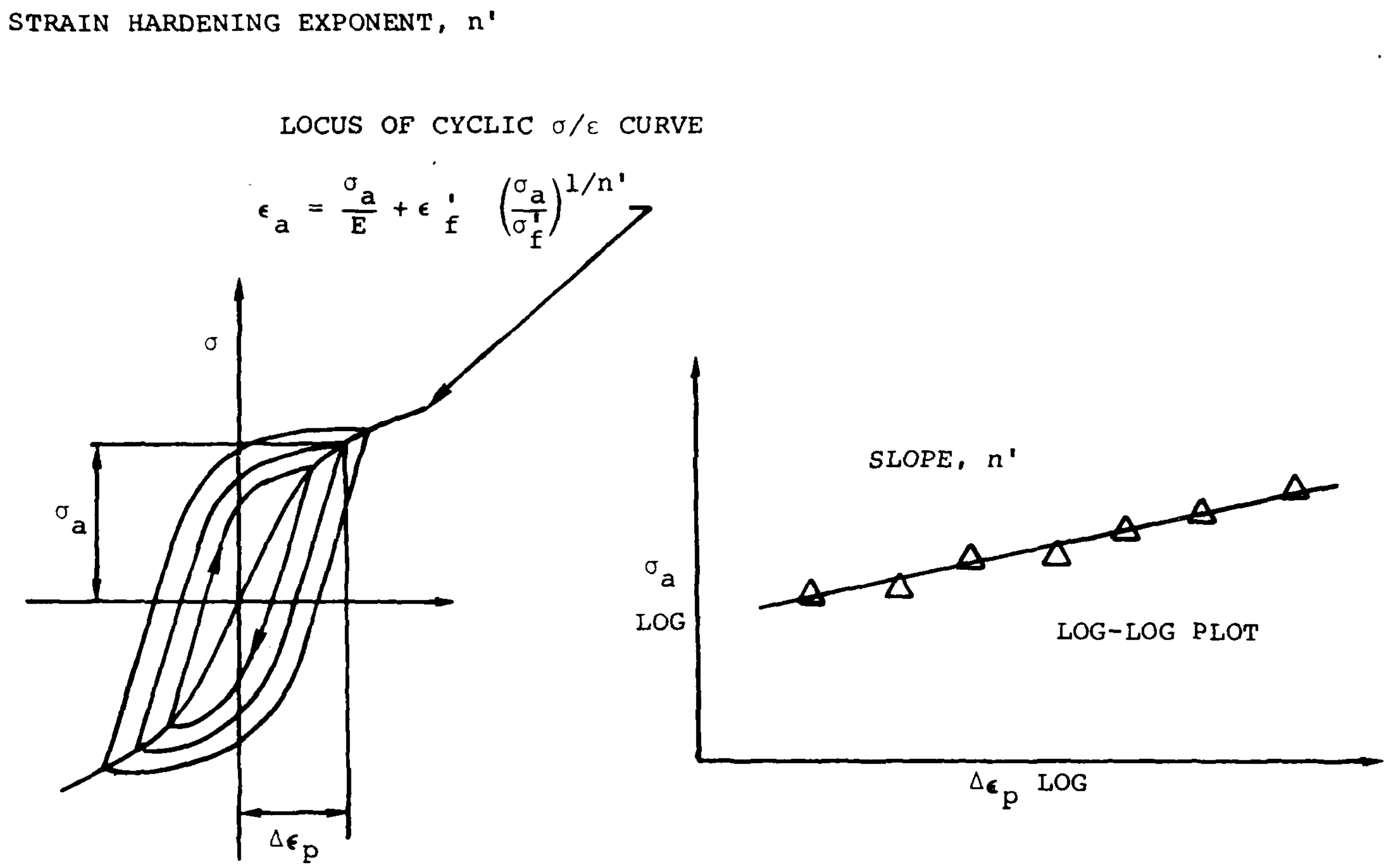


Figure 4-9. Fatigue Life Calculation Method

The acoustic frequencies of the exchanger can be excited by either vortex shedding or turbulent buffeting. So long as their frequencies are within 20 percent of an acoustic frequency, a loud noise is produced.

This acoustic vibration can become destructive when it is in resonance with the natural frequencies of the tubes. During the design, the acoustic frequencies will be changed by inserting, if necessary, sound suppressing baffles parallel to the direction of the airflow to alter the characteristic length.

In keeping with the proposed configuration of the precooler and the intercooler, these baffles, if used, will have a cylindrical shape and extend through the precooler/cooler in an axial direction.

Additionally, the vortex shedding frequency of the airflow as it impinges the tubes is calculated using the techniques developed in "Flow Induced Vibration in Tube Bank Heat Exchangers" Y.N. Chen, Transactions of the ASME, February 1968.

where $f_{n_v} = K \times \frac{K}{D}$ Hz

v = air velocity (ft/sec)

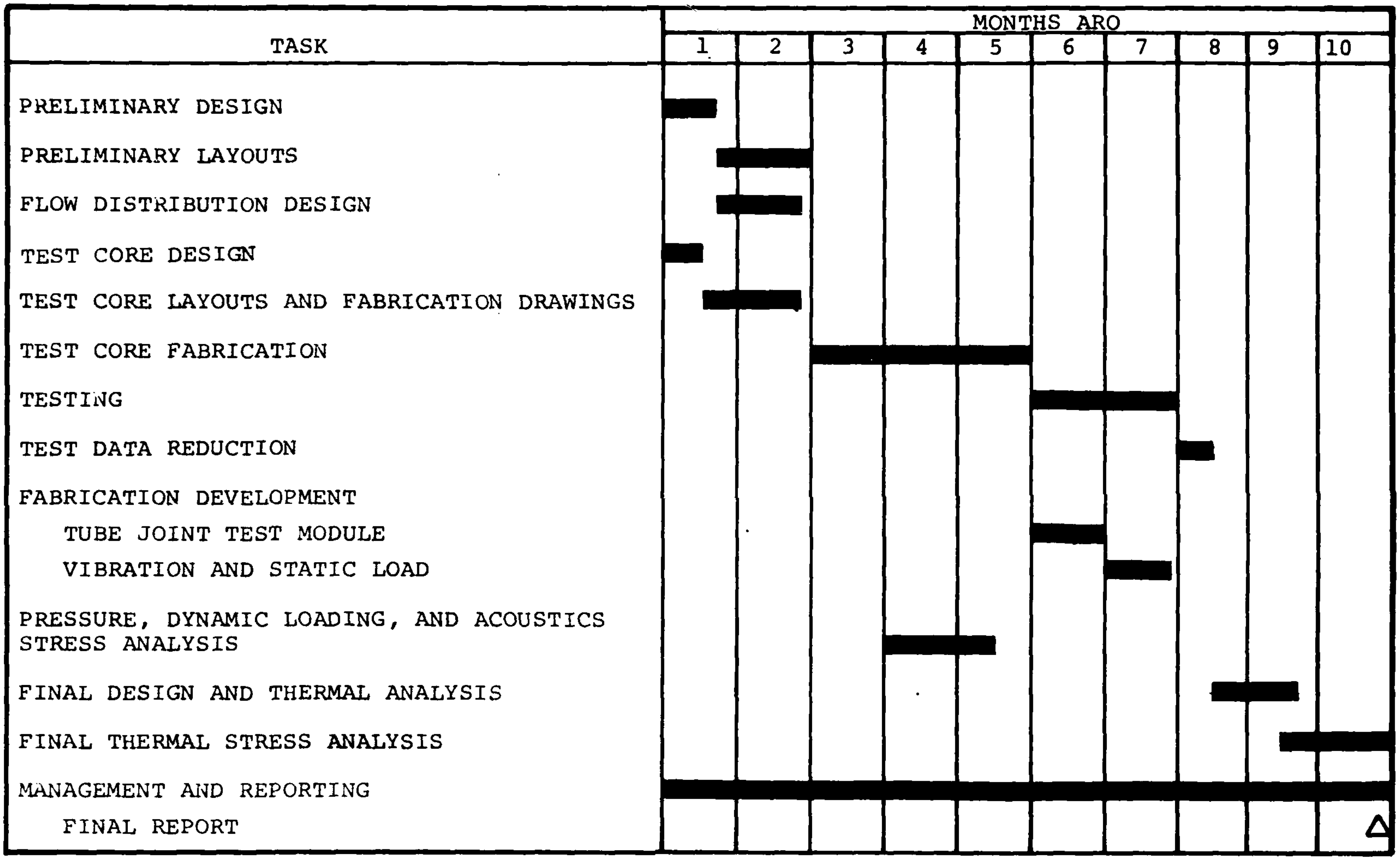
D = effective diameter (in.) of fin tubes

K = function of fluid properties and tube matrix geometry

The effective unsupported length of the fin tubes is adjusted by means of intermediate supports, such that the fin tube natural frequency is greater than the anticipated vortex shedding frequency.

Development Schedule

The schedule for the precooler and cooler development is shown in Figure 4-10. The required development could be accomplished in ten months including the preparation of a final report.



WESTINGHOUSE 70,000 HP CCCBS ENGINE
PRECOOLER AND INTERCOOLER
R&D PROGRAM SCHEDULE

Figure 4-10. Precooler and Intercooler Development Program

4.2.2.2 RECUPERATOR

The proposed recuperator design for the CCCBS is a simple counterflow heat exchanger using a tube and shell configuration. The turbine exhaust helium flows through the tubes while the compressor exit helium flow through the shell. As shown in Figure 7-1, seven of these recuperator modules are wrapped around the turbomachinery. The design is largely conventional, and is based upon the reliable units that are presently in service.

As reported in the Year 1 report, a number of recuperator specimens of a similar geometry to the CCCBS configuration have been assembled, brazed, and tested. Pressure tightness tests were performed both before and after torsional and thermal fatigue testing. No leaks were detected in these tests (Reference 5). In addition, a number of tubular recuperators have been built for European industrial open and closed cycle gas turbines. These designs have demonstrated a high degree of reliability, and in many cases, have run virtually maintenance free for over 100,000 hours of operation (References 6 and 7). The success of these units would tend to reduce the development effort necessary to evolve a satisfactory design.

One of the unique design requirements for the recuperator in the reference CCCBS application is the need to meet the shock specifications of MIL-S-9016 (Navy). However, it is felt that the design effort necessary would be less than that necessary to develop the precoolers and intercoolers.

4.3 MATERIALS

4.3.1 BACKGROUND

In order for development of the CCCBS to proceed, reference materials must be identified for each of the critical components. The reference materials selection must be based on the mechanical properties of the materials and compatibility of the material with the environment in which it must operate. The materials selection process is complicated by the fact that little or no long-term creep-rupture data in the 871°C (1600°F) to 982°C (1800°F) range exists for potential candidate materials even for commercial alloys that have been in service for 10 years or more. Most published creep-rupture data in the 871°C (1600°F) to 982°C (1800°F) range are for test times of less than 1000 hours. The situation is made more complex when materials compatibility with other than air environments are considered. Inert gas working fluids, such as helium, cause a number of problems with respect to their influence on long-term mechanical behavior, particularly on those properties which are sensitive to surface conditions. It has been observed that alloys developed and optimized for use in an oxidizing atmosphere, can experience degradation of mechanical properties when exposed in a nearly inert environment with a very low oxidation potential. In addition, impurity species, such as H_2, CH_4, CO, CO_2 and H_2O are commonly found in inert gas systems. These impurities can also, even when present at low concentrations - $<$500 ppm - have an impact on mechanical behavior of structural materials. The degree of the effect is dependent on a number of factors, namely, the composition of the material involved, the type and concentration of the impurity species involved, the temperature and time of exposure, and the nature of recirculating working fluid system. For example, the ratio of H_2/H_2O and CO/CO_2 determined the oxidizing potential and the ratio of H_2/CH_4 determines the carburization potential in the system. Both potentials influence surface reactions and ultimately may effect material properties. So in order to designate structural materials for the CCCBS with a high degree of certainty that they will perform their design function over the life of system, a great deal of data must be generated to fully characterize the behavior of materials under the anticipated design operating conditions.

In the feasibility study reported in this document, a limited test program was initiated to assess the influence of a helium working fluid environment on the creep-rupture behavior of selected candidate turbine materials. While it was recognized that environment has a significant effect on other mechanical properties such as low and high cycle fatigue, the emphasis of the test program was confined to evaluation of creep-rupture behavior due to practical considerations. The highest combination of stress and temperature occurs in the first stage of the high pressure turbine, thus it was concluded that the turbine design would most likely be limited by creep-rupture consideration. Environmental effects would most likely have their greatest impact at the high temperature-stress condition. In the creep-rupture program reported in the feasibility study, five candidate turbine materials were tested at one temperature - 927°C (1700°F) in air and ultra-high-purity helium. The air and UHP-He test data served as a reference data base. Creep-rupture tests conducted in a dynamic simulated CCCBS helium working fluid were to provide data for assessing the feasibility of turbine materials to function under CCCBS operating conditions. Due to the termination of the program, this phase of the test program was not completed. A limited number of creep-rupture tests were of short duration, less than 3,000 hours. The small amount of data produced prevented the drawing of definitive conclusions. The results were encouraging in that within the scope of time and test conditions no gross deleterious environmental effects were noted for the materials tested. However, the amount of data generated falls far short of what is required to facilitate the further development of the CCCBS.

4.3.2 PROPOSED MATERIALS DEVELOPMENT PROGRAM

The following materials development program is proposed to provide the necessary material property data to facilitate the development of the CCCBS.

- Complete creep-rupture evaluation of initially selected turbine materials in simulated CCCBS helium working fluid. This series of creep-rupture tests was not completed as planned due to the termination of the feasibility study. A limited number of short-term tests were carried out involving two of the five materials under investigation. While gross degradation in the creep-rupture behavior of these materials was not observed, longer term tests of 10,000 hours or more are required to confirm compatibility with the CCCBS helium working fluid.

- Expand creep-rupture test program to include test temperatures in the range 815 to 982°C (1500 to 1800°F) to fully characterize material behavior at "off design" conditions. Stresses at the turbine blade root are expected to be much higher while temperatures are lower than those expected in the airfoil section of the blade.

- Expand creep-rupture test program to include newly developed turbine blade materials in the evaluation process. The MATE program (Materials for Advanced Turbine Engines) sponsored by NASA-LRC has identified a number of potential blade materials with increased creep-rupture strength in the temperature range of interest for the CCCBS. These materials are currently being evaluated for use in aviation gas turbine engines. These newly developed materials offer the same performance advantages to the CCCBS provided compatibility with the helium working fluid can be demonstrated.

- Investigate the effects of simulated CCCBS helium working fluid environment on fracture mechanic properties (crack growth rate), low, and high cycle fatigue behavior of selected turbine materials. These properties are particularly sensitive to surface conditions, thus any modification of the surface of structural components as a result of long-term thermal exposure to a CCCBS working fluid environment could result in a significant departure from expected behavior. These potential effects which cannot be inferred from creep-rupture behavior must be evaluated separately in order to completely characterize structural material behavior in the CCCBS operating environment.

- Investigate the effects of variations in individual containment levels in simulated CCCBS helium working fluid environment on corrosion and creep-rupture behavior of selected turbine materials. During "startup" and "shutdown" operations and at other times during the life of the CCCBS, it is not inconceivable that the level of impurity in the helium working fluid may diverge significantly from steady state levels. The impact of these "off design" conditions on material mechanical property behavior must be investigated and characterized. This can be best accomplished by a systematic study of the effect of variations in concentration of individual and combinations of impurity species on the corrosion and mechanical behavior of selected candidate structural materials. A systematic study would provide a fundamental understanding of the gas-metal interactions involved and thus permit materials behavior to be predicted should actual large CCCBS working fluid composition vary from the anticipated composition.

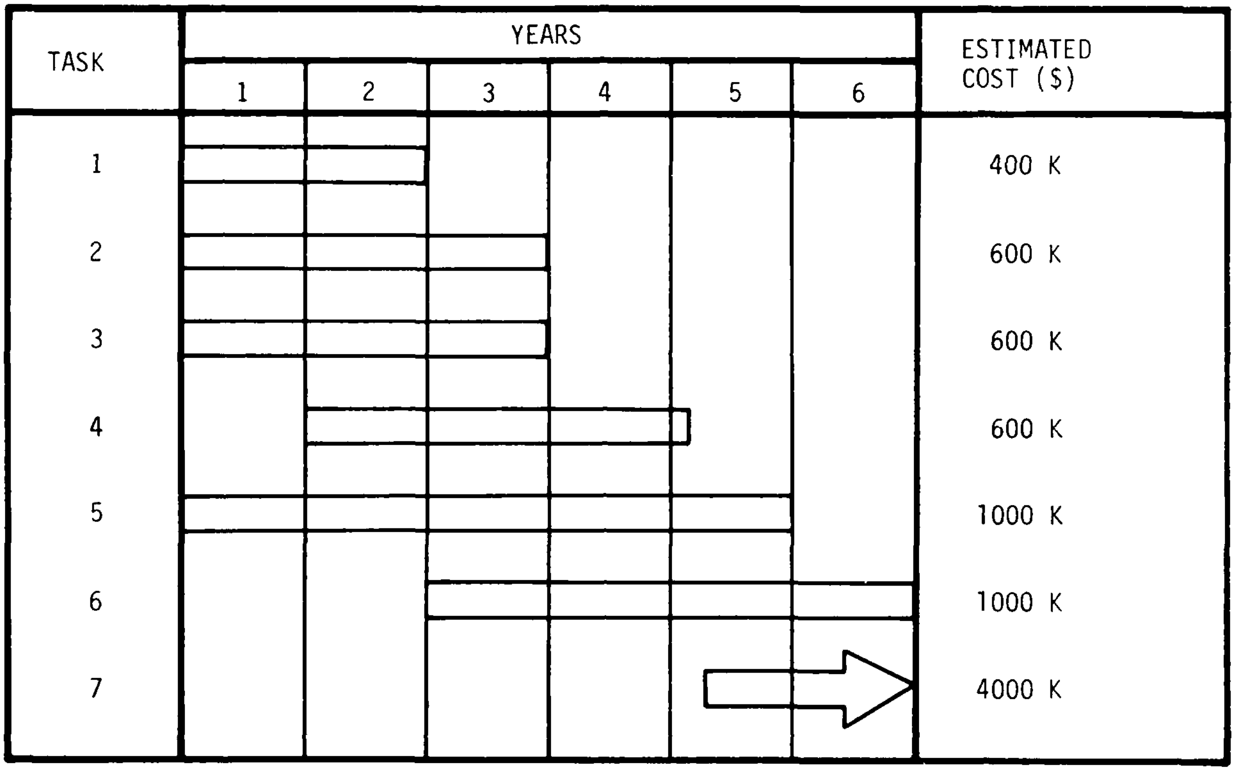
- Conduct alloy optimization research directed specifically toward inert gas environment applications. As stated previously, super alloy development for gas turbine applications has been motivated by the needs of the aviation and land based gas turbine industry.

The primary concern in these applications is compatibility with combustion products; i.e., resistance to oxidation and hot corrosion. The more benign CCCBS helium working fluid should permit a trade-off in alloy composition to meet the requirements of the CCCBS. The alloy optimization study should first look at modification of existing alloys systems for use in the CCCBS environment. However, a full-scale alloy development program beginning with a fundamentally new alloy base may be required. This requirement should become evident in the modification study of existing alloys.

- Generation of Design Data

  Once a reference material has been selected for each CCCBS critical component, based on preliminary evaluations listed above, a testing program can be carried out in conjunction with the detailed design phase and is intended to provide material design limitations under normal operating conditions and under faulted or off design conditions.

MATERIALS DEVELOPMENT SCHEDULE AND COST ESTIMATE



Task 1 - Completion of creep-rupture evaluation of feasibility study.

Task 2 - Expand temperature range of creep-rupture evaluation.

Task 3 - Expand creep-rupture evaluation to include newly developed alloys.

Task 4 - Investigate effects of CCCBS working fluid on fracture mechanics properties - low and high cycle fatigue.

Task 5 - Investigate variation in CCCBS working fluid properties.

Task 6 - Alloy optimization.

Task 7 - Generation of design data.

Figure 4-11. Materials Development Schedule and Cost Data

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5.0 APPLICATIONS OF CCCBS RESULTS

The CCCBS evaluations have been accomplished in the context of what has been judged to be the most stringent naval propulsion application, that of a Surface Effect Ship (SES). Based upon these evaluations, the CCCBS has potential advantages over other types of power plants which make it attractive for other naval applications. In addition, work outside of this program has also indicated applications other than naval vessel propulsion where the CCCBS characteristics make its use attractive. However, this CCCBS program is only one step toward actual availability of a CCCBS power plant and other steps are needed. This section of the report summarizes some of the advantages of the CCCBS that have been identified, discusses some of the applications that can be foreseen, and discusses the next steps that should be taken.

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COMPACT CLOSED CYCLE BRAYTON SYSTEM FEASIBILITY STUDY, VOLUME I--ETC(U)
AUG 79 R E THOMPSON, R L AMMON, R CALVO N00014-76-C-0706
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5.1 APPLICATIONS

The SES was chosen as the representative application for the CCCBS evaluations because that application would place the greatest limitations upon the power plant. In addition, the CCCBS concept definition and evaluations were accomplished within the guideline that the concept must not include features which would exclude other applications. Based upon the results of the design studies, it has been concluded that the CCCBS design concept can be readily considered for numerous applications iwhtout major concept modifications.

The likely applications of a CCCBS are determined to a large extent by the advantages of the power plant concept. Therefore, the more important advantages that have been identified are discussed below, followed by some discussion of likely applications.

5.1.1 ADVANTAGES OF CCCBS

Based upon interpretation of the results of this study, the CCCBS has a number of relative advantages over other power plants being considered for use in high-performance vessels. These power plant advantages would allow the vessels to better meet their mission requirements with less compromises required because of limitations of the power plants.

In a fossil fueled power plant, one of the most important advantages of a CCCBS over other ship power plants is an improvement in the overall plant efficiency (reduction in specific fuel consumption, SFC), especially at part-power operation. As shown in Section 6.1.1, the CCCBS plant efficiency is relatively constant over the normal operating range of the plant. This improvement can provide either an increase in the range of the vessel, or else less fuel and therefore a smaller vessel would be required to obtain the same range as using another type of plant. It should also be noted that the relatively constant specific fuel consumption of partial power that is obtained with the CCCBS is especially an improvement over a normal open cycle marine gas turbine where there is a large increase in the specific fuel consumption at part power. When it is considered that only a small portion of a vessel's operating time is spent at a full power condition, this improvement in part power SFC becomes very important. One mode

of ship operation is to achieve a desired range by operating at the speed which lowers the required power (and hence fuel usage rate) to a level such that the desired range can be achieved. The ship speed reduction that is necessary is magnified with an open cycle gas turbine, due to its characteristics of worsening SFC as power is reduced. The relatively constant efficiency characteristic of the CCCBS can permit achievement of the same range at higher speed, thereby enhancing the mission flexibility of the vessel.

The CCCBS also has the advantage of having a low specific weight and low specific volume. This allows for a compact design and hence, flexibility is allowed the ship designer in the placement of the engines. The relatively small air intakes and uptakes that are required for the fossil fueled energy source and the low exhaust temperature also allow greater flexibility in location of the engine rooms. The ship can be designed to better meet its specific mission requirements with minumum compromises necessary in the ship layout to accommodate the engine.

An additional benefit is the possibility of using a similar power conversion system design with either a nuclear or a fossil energy source. The CCCBS has been conceived so as to be capable of using either a fossil fuel or a nuclear heat source. The number of different types of power conversion systems that would have to be designed and developed for future ships could therefore be reduced, since the power conversion system would require relatively minor changes, and most of the design effort can be directed toward the development of the desired energy source. In addition, the different types of crew training that would be required could be reduced, since the same basic CCCBS could be standard. The port maintenance facilities could also be somewhat standardized. All of these aspects would result in lowering both the initial cost and the operating costs of the ships to the Navy.

The above are advantages that the CCCBS has over other power plants. Since most of the high-performance ships presently under consideration have an open cycle marine gas turbine as a propulsion source, the advantages of the CCCBS over an open cycle gas turbine are of particular interest. Besides the aspects of improved specific fuel consumption, and the ability to use the same engine with

different heat source concepts, there are additional advantages to be obtained by using the CCCBS rather than an open cycle gas turbine.

In an open cycle gas turbine, the flow through the turbine consists of a high temperature mixture of combustion exhaust gases. This gas mixture often results in corrosion and deposition of material on the turbine blades. The CCCBS, however, is a completely closed system with an inert gas such as helium as the working fluid. As a result, the corrosion problems on the plant components are greatly reduced due to the lack of combustion products in the main propulsion engine and the chemical inertness of the cycle helium. This contributes to a longer operating lifetime and lower maintenance requirements than an open cycle gas turbine, also leading to reductions in operating costs.

Besides the use of an inert gas as a working fluid, there are a number of other reasons that lead to an increased operating lifetime for a CCCBS plant. Among them are the fact that there is no ingestion of atmospheric air into the turbomachinery. In an open cycle marine gas turbine, the air intake ducts draw in not only outside air but also other atmospheric contaminants. Among these are salt water, rain, snow, etc., which can lead to corrosion and performance degradation of the engine. The effect of contaminants is reduced in a CCCBS plant, since they only effect the energy source, and as a result longer engine operating times are possible.

As shown in Section 6.1.1, the operating temperatures of the CCCBS are relatively constant over the entire operating range of the plant. This reduces the thermal cycling loads that are felt by the plant, and accordingly help to increase its lifetime potential.

The CCCBS design which was derived in this feasibility study includes small diameter rotating machinery and gas bearings to resist the thrust loads and support the turbomachinery. Very low noise levels are expected from the CCCBS.

As already discussed, the characteristics of the CCCBS allow the engine to be designed for a higher operating life than a normal open cycle gas turbine. Part of the design philosophy applied in the design of the CCCBS was to reduce as much

as possible the required maintenance on the engine. The entire propulsion plant (turbomachinery, recuperator, precooler and intercooler) can be assembled into a more compact package than would otherwise be possible. Due to the design for high reliability of the system, normal maintenance is required only on those components external to the turbomachinery/heat exchanger package. This allows the ship designer greater freedom in the placement of the engines, since there is no need to require a large amount of space around the engines to enable a multitude of maintenance and repair operations to be carried out. Also, due to the design for high reliability of the CCCBS, there is the added potential of reducing the crew size needed. This would be possible since the engine room crew could be reduced because of the lower maintenance requirements on the engine with an important impact on reduction of operating costs.

5.1.2 APPLICATIONS

Indications are that the large SES continue to be one likely application for a CCCBS power plant for several reasons. Improved SFC, particularly at partial power is of major importance in that application because of the high power levels required for the ship and because of its limitations on the weight of fuel that can be carried. In addition, several other characteristics of a CCCBS power plant appears to be important to a large SES design.

The additional flexibility in engine room location that a CCCBS allows can be of importance in some SES. The Reference 1 study to determine a minimum size SES carrier is an example. In that study the hangar deck essentially sized the ship. The hangar deck dimensions were determined by the space required for the aircraft plus the space required for the six LM-5000 gas turbine engines which had to be located on the hangar deck. It appears that the smaller air intakes and uptakes required for a closed cycle gas turbine power plant and the compactness of a CCCBS would permit the CCCBS power plant engine rooms to be located below the hangar deck. In that case, the hangar deck dimensions, and therefore also the overall ship dimensions, could be reduced thus realizing important savings.

The Reference 1 SES study also included one SES configuration powered by two conceptual Light Weight Nuclear Powerplants (LWNPs) each of which utilize two compact closed cycle gas turbines similar to the CCCBS. Reference 1 showed that

the nuclear powered SES would be 50 feet (10%) shorter, primarily because of the smaller space needed on the hangar deck for the LWNPs and 22% lighter.

The preliminary CCCBS results and CCCBS characteristics were porvided to Bell Aerosystems as one input to the study of prime movers and thrustors for large SES that they are conducting for the SES Project Office. Subsequent consultations with Bell Aerosystems personnel have indicated that the CCCBS has characteristics that warrant its inclusion as one of the prime candidate power plants for large SES of the future.

Other naval vessel types for which CCCBS power plants will be candidates essentially include all those for which open cycle aircraft derivative power plants are candidates. The applications are expected to include such as large hydrofoils, destroyers and frigates, and Small Waterplane Area Twin Hull (SWATH) ships. In these installations, the SFC improvement realizable with a CCCBS power plant will be of importance. In the case of large SWATH, the possibility of locating the CCCBS power plants in the submerged hulls may also be an advantage.

It has also been determined in other studies that the CCCBS technologies can be appropriate for other than naval propulsion application. For instance, References 2 and 3 documented the results of a design and evaluation study of a new concept for utilizing a variant of the closed cycle gas turbine in a system for nuclear aircraft propulsion. It was determined that the "Bi-Brayton" system would provide many advantages and would significantly simplify some of the propulsion system design features if nuclear powered aircraft are determined to be needed by the Air Force or Navy.

The Reference 2 study for the Air Force directly benefitted from this CCCBS research in the areas of turbomachinery, compact gas-to-gas heat exchanger, and system design technologies. The general configuration and sizing of the Bi-Brayton turbomachinery was developed by appropriately scaling from similar CCCBS components, thereby also demonstrating the applicability of CCCBS technologies to this application.

It is also interesting to note that the Bi-Brayton system variation of a CCCBS might have application to naval vessels which require that power must be transmitted throughout the ship (such as in large SES). Fundamentally, the Bi-Brayton system concept locates the power turbine at the location of the load while the energy source and gas generator are more centrally located. Only moderate and low temperature heat exchangers and gas transmission lines are required, thus making remote power turbine location realistic.

Another candidate non-naval application for a CCCBS is that of a solar central station power plant. In this application, the small size and weight of a CCCBS could allow its location near the top of a power tower thereby minimizing the piping required and simplifying the system.

5.2 REFERENCES

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2. "Nuclear Bi-Brayton System for Aircraft Propulsion Study - Final Report," Contract No. F33615-77-C-0116; Westinghouse AESD; WAES-TNR-234; March 1978.

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